PERFORMANCE OF VORTEX CONTROLLED HYBRID DIFFUSERS WITH NON–UNIFORM INLET FLOW

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CONTENTS

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ABSTRACT

A conventional diffuser of a gas turbine engine located between the compressor and combustion chamber has a pre-diffuser followed by a dump diffuser. A short hybrid diffuser with a boundary layer bleed capability has considerable advantage over such conventional diffusers in terms of effectiveness, length and control of exit velocity profiles. Since the compressor exit velocity profile is distorted experimental investigations were carried out on such short diffusers with symmetrically and asymmetrically distorted inlet velocity profiles for area ratio 2 and 2.5 and divergence angles of 30^{0} and 45^{0}. For each of the above configuration experiments were carried out by varying the fence subtended angles from 0^{0} to 30^{0} and bleed rate varying from 0% to 8%. The performance was evaluated in terms of diffuser effectiveness, vortex chamber depression and quality of exit velocity profile. Experiments were

also carried out with differential bleed to get desired exit velocity profiles. Flow visualisation with kerosene vapour and Titanium-tetra-chloride was attempted. Investigations were also done on the performance of the diffuser with vortex generators.

NOMENCLATURE

A	Cross sectional area
AR	Area ratio, expressed as a ratio of exit to inlet area
B	percentage of main flow which is bled out to vortex chamber
B_1	Dimension of the square primary duct
B_2	Cross sectional width at the base of vortex control step in the case of hybrid diffuser.
B_3	side dimension at the exit of diffuser
C_d	Coefficient of discharge
C_p	Coefficient of static pressure recovery
h	In graphs, it is the height of the duct in which the experiment is carried out.
L	Length of secondary duct
n	No. of traverse points having an equal area weighting
P	Static pressure
P_1	Static pressure at the entrance of the diffuser
P_2	Static pressure on the walls of the secondary duct
Q	Volumetric flow rate in m^3/sec.
Re	Reynold's number.
u	Velocity in the primary duct
u_{av}	mean or mass average velocity
u_{max}	maximum velocity
V_c	Vortex chamber depression
x	axial distance between primary duct and fence.

y	vertical distance between the fence and the inner surface of primary duct.
α	kinetic energy flux parameter(defined in text)
ϕ	fence subtended angle
θ	diffuser angle (see fig.1 also)
w	width of the secondary duct
η	Diffuser effectiveness (defined in text)
ρ	Density of air at test conditions.
V.G	Vortex generators.

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CHAPTER-1

INTRODUCTION

In a gas turbine engine the high velocity gases from the compressor are at first decelerated in a pre-diffuser and then in a sudden expansion dump diffuser before entering the combustion chamber. This conversion of kinetic energy into stream pressure is required for the jets to penetrate the dilution and primary ports. Failure to achieve this can have detrimental effects on combustor exit temperature distribution,emission levels and liner integrity. But such diffusers exhibit high losses due to dumping of gases in the dump region. Fishenden and Stevens (1) found that strong seperation was inevitable if such diffusers were made short. However diffusers cannot be long due to increase in weight and also mechanical complication that may arise due to the distances imposed by the diffuser between the compressor spools and their driving turbines. Hence short diffuser with a low pressure loss would be befitting in a gas turbine combustor. Since a short diffuser is very much prone to seperation it has to be controlled by some method which can energise the boundary layer. This can be achieved by means of vortex generators, injection of high velocity fluid in the boundary layer by means of tangential blowing or by means of boundary layer suction. Seperation can also be avoided or delayed by distributing the stream lines with the help of splitter vanes. Boundary layer suction in a wide angle diffuser through a chamber creating a vortex which imparts momentum to the slow moving fluid in the boundary layer, can give a good pressure recovery and at

the same time enable to monitor the exit velocity profile.

Concept of Vortex Controlled Diffuser

The concept of vortex controlled diffuser was first described by Ringleb (2). He pointed out that by aerodynamic design of cusps in a short diffuser, the vortices formed in the direction of flow can overcome the strong adverse pressure gradient. However maintaining a stable vortex would be very difficult as energy would be lost within cusps by skin friction. Later on vortex controlled diffuser proposed by Adkins (3) could overcome this problem by bleeding off some percentage of main flow through a seperate chamber just at the exit of the primary duct (i.e near the throat of the diffuser) and maintaining a stable vortex. This component is called the vortex chamber (Fig.1). The mechanism of maintaining a stable vortex is shown in Fig.2. Due to suction the static pressure in the vortex chamber would fall and hence stream tube 'a ' is drawn into the vortex chamber and it decelerates as it approaches the chamber enterance. The stream tube 'b ' continues to flow through the diffuser and decelerates (as it is moving into a region of greater static pressure) appreciably in the region of vortex chamber. As a result of this a highly turbulent shear layer is created resulting in energy transfer from stream 'a' to stream 'b'. Consequently some of the fluid in stream 'b' which has less energy for further diffusion is thus able to flow through the diffuser without developing any stall. At the downstream of the throat another vortex is formed in the direction of flow due to the formation of the "Coanda bubble" which also helps in avoiding

or delaying seperation. In the case of hybrid diffusers Fig.6 (4) vortex controlled step accounts for a small increase in area and would therefore not experience a very strong pressure gradient. Experiments were conducted on hybrid diffusers for angles 30^{0} and 45^{0} and area ratio 2 and 2.5 with various inlet velocity profiles and fence subtended angles.

REVIEW OF LITERATURE

Considerable attention was given to study the literature regarding control of diffuser outlet profile and the performance of combustor at various exit profiles. Fishenden and Stevens (1) conducted experiments on Annular combustor-dump diffusers and found that combustor performance depended on the dump gap distance (distance between flame tube and outlet of the diffuser),split ratio(ratio of mass flow through outer annulus to that of inner annulus) and dimensions of pre-diffuser(area ratio and length). Experiments led them to the conclusion that for a particular pre-diffuser ther was a narrow range of split ratio and dump gap distance for which the flow at the outlet of the diffuser was unseperated. Also they noted that highly distorted radial velocity profile at the exit of the pre-diffuser created asymmetric pressure distribution at the head of the flame tube resulting in local flow reversal and thereby causing over heating of the flame tube. Since tests carried were for a fully developed flow at the inlet,the set of experiments failed to reveal the performance of such diffusers at unsteady wake profiles usually encountered at the exit of an axial flow compressor. Influence of blade wakes on the performance of

combustor shortened pre-diffusers was examined by Stevens,Harasgama and Wray (6). They noted that wakes decay or grow depending on relative magnitude of pressure and shear forces. During the initial stages of diffusion the wakes actually helped in overcoming the adverse pressure gradient due to high level mixing. Only small decrease in total pressure was noted. When the outlet guide vanes were sited very close to the entrance of the optimum length diffuser there was an improvement in stability of the outlet velocity profile as compared to that of fully developed flow. Later on Klein(7) carried out further experiments to find out relation between losses and entry flow conditions in such short dump diffusers. Cascades of compressor blades up-stream of the diffuser were used to make the flow at the inlet similiar to that in a jet engine. The flow field was altered by varying the distance between the cascades and the diffuser inlet plane and by changing the blade aspect ratio. The measurements clearly showed that distortions in radial direction affected the losses to a much larger extent than non-uniformities in the circumferential direction.

Adkins (3,4) worked on short diffusers for gas turbine combustors because of its inherent advantages in an aircraft engine. Mild suction resulting in the formation of a stable vortex helped in improving the effectiveness drastically. He has tried to correlate the measurements taken for area ratios in the range of 1.9:1 to 3.2:1. Adkins inferred that a pressure coefficient in excess of 80% may be obtained with diffuser lengths only one third of optimum conventional diffusers using a bleed off rate of approximately 5% of the main flow at the

diffuser throat. Fletcher and Adkins (5) have used the concept of this vortex controlled diffuser in tandem with the combustion chamber in investigations related to control of pollutant emissions from gas turbine combustors. They used the bled vortex generated to control and vary the air distribution with in the combustor liner. They inferred from the experiments that pollutant emission reductions can be achieved by use of variable geometry. Albert J.Juhasz(8,9,10,11) used the same vortex control principle in short annular diffusers. However in addition he succeeded in varying the exit profiles by differential suction from outer and inner annuli. He observed that suction rate beyond a particular value from only one wall resulted in seperation at the opposite wall. However mild suction rate from the opposite wall helped to prevent seperation.

Senoo and Nishi (12) conducted experimental investigation over a set of diffusers having divergence angle in the range from 8^{0} to 30^{0}, for an area ratio of 4 using vane type vortex generators and a tail pipe. It was stipulated that trailing vortices from vortex generators provided an induced component perpendicular to the wall to delay or prevent seperation in an adverse pressure gradient. By using vortex generators,flow fluctuations of wall pressure was reduced for divergence angle less than or equal to 16^{0}. But above 16^{0}, even with the usage of vortex generators, flow was unstable and pressure fluctuations were large. Introduction of tail pipe seemed to have improved the pressure recovery. They inferred that for thick boundary layers at inlet, there would be provision to

improve co-efficient of static pressure recovery with vortex generators.

Bettner(13) did experimental investigations on turbine stator blading. He adopted some boundary layer control concepts like vortex generators and tangential blowing. He investigated on two types of vortex generators - two dimensional co-rotating vane vortex generator and three dimensional triangular plow vortex generators. The objective behind this investigation was to use a mixing process as the mechanism of boundary layer re-energization. The vortices shed from the above devices was to promote mixing of the high energy free stream flow with the retarded and low energy boundary layer flow.

<u>Literature survey for velocity profile generation</u>

Wen and Jienkiervicz (14) obtained by experiments a nearly uniform shear flow in a working section of a wind tunnel by inserting a grid of parallel rods placed at variable spacing. The function of such a grid was to impose a resistance to the flow so graded across the working section as to produce a linear variation in velocity profile. Studies on steady two-dimensional flow through an arbitrary shaped gauzes of non-uniform properties placed in a channel was done by Elder (15). Lively and turner (16) modified the non-uniformity of upstream flow that seemed to have existed in the work of Wen and Jienkiervicz (14) by improved grid spacing. They applied the method of generation of linear flow successfully to the case of symmetrical velocity profiles in two dimensional duct. They have also discussed about the decay of the velocity profiles. Turner (17) presented a computational model for the analysis of two dimensional flow through non-

uniform gauzes. His method was entirely based on the linearised theory of Elder, but numerical techniques were relatively simple. Boundary layer thickness at the exit of the multistage axial compressor was around 20% of the duct size near both the walls and the velocity profiles did not deteriorate in succeeding stages, after stage four as in Ref.18.

Scope of the present work:

A diffuser is an essential component between the compressor and the combustion chamber for the gases at the exit of the compressor to be decelerated with low losses so as to keep the specific fuel consumption low and also to establish a high static pressure for effective jet formation. The vortex controlled hybrid diffuser satisfies both these requirements with only one-third the length of an optimum conventional diffuser. Moreover the exit velocity profiles could be altered according to the engine requirements by differential suction from the top and bottom walls. Such a monitoring of exit velocity profiles can be very useful during idling operation when a hub or tip peaked velocity profile is needed so as to by-pass most of the flow and during cruise condition when a center peaked velocity profile at the inlet of the combustor is required. Thus it has got an advantage over conventional dump diffusers in terms of pressure recovery, length and quality of exit velocity profile.

Present experimental investigations have been carried out in two dimensional hybrid diffusers, in view of facilities and time scale involved. The investigations done by Adkins (4,5) and Albert J.Juhasz (8,9,10,11) were on annular type and in the case of hybrid diffusers it was for conical type. Dolan and Runstadler (20) have compared the performance and

observed only a slight fall for square inlet diffusers as compared to conical diffusers. Therefore, the present measurements on rectangular diffusers were expected to provide a conservative estimate of performance for conical or annular diffusers and could be compared with the results of Adkins and Albert J.Juhasz(3,4,9).

Experiments were carried out on hybrid diffusers for area ratio 2 and 2.5. Initially experiments were carried out with symmetric inlet velocity profiles and later on with asymmetric (hub-peaked) velocity profile. For each area ratio the bleed rate and fence subtended angle was varied to obtain the combination of optimum bleed rate and effectiveness. All these experiments were carried out for a Reynold's number of 10^5. (Reynold's no. based on primary duct dimensions). Inlet velocity for most of the experiments was 20 m/s. Bleed rate was varied from 1% to 4% . Readings were taken for no suction case also. After findng out that for a distorted inlet velocity profile the outlet profile is also distorted in the same fashion for uniform suction from top and bottom walls , differential suction was employed to make the outlet velocity profile uniform. Flow visualisation was attempted to understand the complex nature of the flow (Adkins (3) has given only a schematic representation of the flow in the vortex chamber) .Initially flow visualisation was attempted with kerosene vapour and when it failed to yield positive results ,visualisation with Titanium tetrachloride was tried. Towards the end investigations were done on hybrid diffusers with vortex generators with a view to minimize suction or possibly eliminate it completely. The vortex generators were designed with help of Ref.12.

The results presented from the present investigations include the set of performance curves for hybrid diffusers for symmetrical velocity profile (U_{max}/U_{av} =1.14) and asymmetric velocity profile (U_{max}/U_{av} =1.17) which was hub-peaked. Hybrid diffusers with area ratio 2 and 2.5 and angle of divergence, 30 and 45 degrees each were tested for various bleed rates and fence subtended angle and for a particular inlet velocity profile. Turbulence measurements were taken at inlet and exit of the diffuser.

CHAPTER-2

EXPERIMENTAL SET UP

The apparatus consists of a small tunnel mounted on an adjustable platform. As shown in fig-4,the tunnel layout consists of a flexible air supply pipe,a settling chamber,contraction,primary duct,vortex chamber and secondary duct. Compressed air bottles of the trisonic tunnel with a capacity of 85 cu.m at ambient temperature and a maximum pressure of 18.7 kg/sq.cm supplies air to the tunnel (fig 5) after regulating the flow through a set of pressure reducing valves and pnuematically operated butterfly valve .

DIFFUSER DESIGN

Experiments conducted with similiar diffusers had been done with a Reynold's number of around 10^5 (ref-3,4).

The cross section of the primary duct made of perspex is 71.5 mm X 71.5mm and has a thickness of 6mm. It has a length of 560 mm i.e about 8 times the side dimension so as to impose suitable velocity profiles generated by the grid. The dimensions of the secondary duct are 71.5 mm X 143 mm and 71.5 mm X 178.7 mm for area ratios of 2.0 and 2.5 respectively. Alumunium plate fences of 130mm X 95mm X 3mm were clamped between the flanges of primary and secondary ducts. The fence had slots on sides to adjust its location. Vortex controlled step ratio was taken as 1.2:1 for the hybrid diffuser (ref 4). Experiments were carried out in diffusers for two divergence angles 30^0 and 45^0 whose values are tabulated below.

Table 1

diffuser inluded angle	B_1	B_2	B3 AR=2	B3 AR=2.5	Inclined length AR=2	Inclined length AR=2.5
30^0	71.5	86	143	178.7	112	180
45^0	71.5	86	143	178.7	76	122

All dimensions in mm

A tail pipe of length 130mm was also connected to the secondary for better diffusion.

A suction system with a capacity of 102 cum/hr was employed to enable the boundary layer to be bled off equally from the top and bottom of the junction of primary and secondary duct through the vortex chamber. The suction pump is capable of handling bleed off rates up to 8% of the main flow for primary duct velocities in the range of 20 to 25m/secs. Flow rate is measured using orifice meter and bleed rate is controlled by a butterfly valve. Photograph 2 shows the suction system and fig 7 the schematic details.

Vortex chamber is the central part of the diffuser for it is through this chamber that a fraction of the main flow is bled to create a vortex which is instrumental in the prevention of seperation (fig 8). Air entry to the vortex chamber is governed by the vortex fence subtended angle which is one of the most crucial parameters affecting the performance of the diffuser. It can be varied by changing the 'x' and 'y' distance (fig 6). However during the course of experiments for convenience sake the 'y' distance was kept fixed and 'x' distance

alone was varied. Experiments were conducted with fence subtended angles ranging from 0 to 30.

COMPRESSOR OUTLET FLOW SIMULATION

Seldom is the velocity profiles at the outlet of an axial flow compressor uniform and hence diffuser tests has to be carried out with distorted velocity profiles which was indeed done with centre peaked velocity profile ($U_{max}/U_{av}=1.14$) and later on with a hub skewed velocity profile ($U_{max}/U_{av}=1.17$). Generation of these velocity profiles were done with help of grids following the procedure given in ref 14 to 17. Initially grid tubes (1mm dia stainless steel)were spaced so as to generate nonuniform flow up to 16% of the duct height on both top and bottom walls. Subsequently an asymmetric velocity profile usually encountered in compressor exit flows was generated by placing grids on one side up to 65% of the duct height. This resulted in a hub skewed velocity profile with maximum velocity occuring at a height of 35% from the bottom wall.

DIFFUSER INSTRUMENTATION

The pressure difference across the orifice was measured using a multitube manometer with an accuracy of 1mm of water. A Dweyer gauge with a range up to 7.5cm of water was used for wall pressure measurements. This manometer had higher accuracy in the lower range. An electronic digital manometer with a barocel differential pressure sensor of accuracy 0.025 mm of water was also used for some of the wall measurements and for dynamic head measurements . Velocity distribution in the primary and secondary was taken with the help of a 1mm dia pitot tube mounted on a three dimensional traversing mechanism. The traverse was accurate

to an order of 0.05mm in vertical direction and 0.6mm in horizontal direction. Photograph 3 shows some of the instrumentation and traverse mechanism. DISA constant temperature hot wire anemometer was used to measure the turbulence distribution in the diffuser primary and secondary ducts.

Flow visualisation was attempted with help of kerosene smoke generated by the smoke generator and also with Titanium tetra chloride. Later on experiments were carried out with vortex generators. Vortex generators were designed on the basis of ref.12. Fig.32a and 32b shows the dimensions and location of the vortex generators respectively. The vortex generators were soldered to a very thin plate of thickness 0.5mm

CHAPTER-3

TESTING TECHNIQUE

The measurements involved in the present set of experiments were velocity, pressure and turbulence measurements. The flow rate through the primary could be varied by means of actuating the butterfly valve which was monitored by pressure regulator located besides the settling chamber (Fig.5). The velocity in primary duct in most of the experiments were maintained at 22.5m/s. At first velocity traverses were taken to ensure that grid placed at the inlet of the primary did generate the required velocity profile. The velocity measurements were done with the help of pitot-tubes and Dweyer gauge and traverses by means of traversing mechanism as shown in photograph-3. Turbulence measurements at the exit of the primary was done with the help of DISA constant temperature anemometer.

Pressure and velocity measurements were taken at the secondary of each hybrid diffuser for various fence subtended angles and bleed rates. The fence subtended angles were adjusted by means of varying the 'x' gap while maintaining the 'y' gap at 3.4mm (Fig.6). The suction system (Fig.7) bled off equal amount of air from both the walls. The amount of air bled off was controlled by means of butterfly valve and was measured by means of orifice meter.

Static pressure measurements were taken on diffuser walls for both top and bottom surfaces. A pressure tapping was located at 71.5mm (one side dimension of primary duct) ahead of primary duct exit to serve as a reference static

pressure reading. The pressure tapping locations in the secondary duct are shown in Fig.10. More tappings are provided after 30^0 position where the pressure recovery was expected to be more. One tapping is provided in both the vortex chambers for measuring the value of vortex chamber depression.

Velocity traverses for any particular configuration were carried out at the centre and at locations quarter width from the walls.

As mentioned earlier Flow Visualisation with kerosene vapour and Titanium Tetra Chloride was tried. A hole was drilled just before the exit of the primary duct and kersene vapour was introduced from the smoke generator by means of a tube. Various smoke injection velocity and density was tried but still the smoke failed to flow the stream lines of the vortex. Hence viualisation with the help of Titanium Tetra Chloride was tried. The smoke although denser compared to kerosene vapour still failed to follow the stremlines , instead got diffused immediately.

Towards the end performance of the diffuser with vortex generators was investigated. Fig.32a and 32b shows the dimensions and location of vortex generators (Ref.12). The effectiveness of the diffuser with and with out suction with vortex-generator located at 0.4 times the primary duct height upstream of the secondary duct inlet was found out. Later on the vortex generators were shifted further upstream by double the original distance and the performance was investigated.

CHAPTER-4

PERFORMANCE CALCULATIONS

The overall performance of a diffuser is usually evaluated in terms of total pressure loss, effectiveness, and quality of exit velocity profile.

Law of conservation of mass for steady flow gives

$$\dot{m}_1 - \dot{m}_b - \dot{m}_2 = 0 \qquad (1)$$

$\dot{m}_1$ = Inlet mass flow rate

$\dot{m}_b$ = Bleed rate

$\dot{m}_2$ = Exit mass flow rate

Defining $B = \dot{m}_b/\dot{m}_1$, the continuity equation governing the flow can be written as

$$\rho A_1 u_{av1}(1-B) = \rho A_2 u_{av2} \qquad (2)$$

giving

$$u_{av2} = \frac{(1-B)u_{av1}}{AR} \qquad (3)$$

u_{av1} = inlet average velocity

u_{av2} = outlet average velocity

AR = area ratio

Defining a parameter called kinetic energy flux parameter as the ratio of mass flow weighted dynamic head to mass averaged dynamic head to take into account of distortion of velocity profile the Energy equation gives

$$p_1 + 1/2\, \alpha_1 \rho u_{av1}^2 = p_2 + 1/2\, \alpha_2 \rho u_{av2}^2 + \text{losses} \qquad (4)$$

where α_1 = kinetic energy flux parameter at inlet

α_2 = kinetic energy flux parameter at outlet

ρ = density of air at experimental condition

p_1 = static pressure at inlet

p_2 = static pressure at outlet

The quantity α is defined as

$$\alpha = \frac{n^2 \sum_1^n (\sqrt{h})^3}{(\sum_1^n \sqrt{h})^3} \qquad (5)$$

where h = measured dynamic head

n = number of traverse points having equal area weighting

The value of n taken was at least 20 in all the experiments. Velocity traverses were taken at three vertical and horizontal stations particularly at the diffuser exit. Thus the ideal pressure rise can be calculated using the expression

$$P_2 - P_1 = 1/2\, \rho\, u_{av1}^2 \left[\alpha_1 - \alpha_2 \, (u_{av2}/u_{av1})^2\right] \qquad (6)$$

Diffuser effectiveness defined as measured pressure rise divided by ideal pressure rise can be calculated as given below

$$\eta = \frac{(P_2 - P_1)_{measured}}{1/2\, \rho\, u_{av1}^2 \left[\alpha_1 - \left(\frac{1-B}{AR}\right)^2\right]} \qquad (7)$$

The value of P_1 was measured one hydraulic diameter (in the present case of square duct it is one side dimension) upsytream of vortex chamber inlet. For hybrid diffusers, P_2 was measured at the exit of the diffuser i.e at the enterance of the tail pipe. The pressure lost by the bleed air has been expressed as the non-dimensional vortex chamber depression, v_c where

$$V_c = \frac{P_1 - P_c}{1/2\, \alpha_1\, \rho\, u_{av1}^2} \qquad (8)$$

Vortex chamber pressure P_c was monitored from top and bottom suction chambers. It was ensured that this pressure was same for both the chambers as would be required for equal suction from both the sides. In case of minor variations, an average of the two readings was taken.

CHAPTER-5

RESULTS AND DISCUSSIONS

Shantaram, K.V (19) had done investigations with uniform inlet velocity profile. Since the compressor exit profiles are non-uniform, the present investigations were carried out with symmetrically distorted inlet velocity profile (u_{max}/u_{av}=1.14) and asymmetrically distorted inlet velocity profile (u_{max}/u_{av}=1.17). At first studies were carried out with symmetrical velocity profile generated by the grid (Fig.8). The turbulence intensity was also measured at the exit of the primary duct (Fig.12). Turbulence intensities obtained were of the order of 3 - 4% . This shows that placing the grid has not affected the turbulence level. Fig.11a, 11b shows the velocity profile generated by the grid.

RESULTS FOR SYMMETRIC INLET VELOCITY PROFILE

Fig.13 shows the effect of suction and fence subtended angle on the pressure recovery of the diffuser. The figure shows that at lower bleed rates, fence subtended angle of 25^{0} gives the best performance for a diffuser with AR=2 and divergence angle 30^{0}. This could be because the combination of radial and axial gap corresponding to this angle enabled a stable vortex to be formed at lower suction rates.For fence subtended angles of 20^{0} and 30^{0}, higher suction rates are required to achieve similiar diffuser effectiveness. As high diffuser effectiveness is desired at lower suction rates, fence subtended angle, of 25^{0} is optimum for this area ratio and divergence angle. For comparison with other diffuser configurations, results for optimum fence subtended angle were used.Fig.14 shows the effect of non-uniformity of

inlet velocity profile on the effectiveness of the diffuser. It is seen that uniform exit velocity profile gives better effectiveness because of a thinner boundary layer. Thicker boundary layer of the non-uniform inlet velocity profile needs a high momentum transfer to the bulk of low kinetic energy fluid overcoming the strong adverse pressure gradient, for which larger suction rates are required as seen from the figure. Fig.15 shows the variation of diffuser effectiveness along the length of secondary duct. It is seen that that tail pipe improves the pressure recovery at all bleed rates. It was also noted that the velocity profile tends to become uniform downstream in the tail pipe. Enhanced mixing is the reason for improvement in pressure recovery. Fig 16a and 16b shows the velocity profiles along vertical (u_{max}/u_{av}=1.6) and horizontal (u_{max}/u_{av}=1.27) axis for area ratio 2. Fig 17a and 17b shows the vertical (u_{max}/u_{av}=1.92) and horizontal (u_{max}/u_{av}=1.42) velocity profile for area ratio 2.5. From the above figures it can be inferred that u_{max}/u_{av} which is a measure of distortion is more along the vertical axis, also the level of distortion is more for area ratio 2.5 because of higher losses. Fig.18 shows the exit velocity profile along the vertical axis for divergence ange of 45^0 and area ratio 2.5. It is seen that due to higher divergence angle the pressure gradient to overcome is more and hence losses are higher which is clearly reflected as higher distortion in the exit velocity profile (u_{max}/u_{av}=2.11) as against 1.92 for 30^0 divergence angle. It is to be noted that velocity traverses were conducted at w/4 distance (w is the width of the duct) from the left and right walls along the vertical axis and h/4 distance (h is the height of the duct) from the top and bottom walls along the horizontal

axis. In almost all cases the variation in velocity profile distortion when taken at different stations at the plane of the secondary exit (inlet to the tail pipe) was not substantial. Fig.19a to 19f shows the velocity profiles along horizontal and vertical axis taken at quarter height and quarter width respectively from the walls. It is found that distortion in vertical velocity profile varies from 1.52 at the centre to 1.88 near the left wall and 1.69 near the right wall. The horizontal velocity profile varied from 1.22 at the centre to 1.17 near the top wall and 1.26 near the bottom wall. Fig.20 shows the effect of suction on vortex chamber depression. The figure shows higher vortex chamber depression at higher bleed rates showing larger pressure loss of bled air at higher suction rates. Fig.21 shows higher area ratio gives lower effectiveness although to a smaller extent, owing to higher losses. From Fig.22 it can be inferred that higher divergence angle gives lower effectiveness. This is due to the fact that higher divergence angle imposes a larger pressure gradient and hence suffers greater total pressure loss. However looking at the gas turbine engine as a whole, the performance may be better when incorporating a higher divergent angle diffuser owing to the reduction in weight and length.

RESULTS FOR ASYMMETRIC VELOCITY PROFILE

In the second stage the performance of the diffuser was tested with an asymmetric hub-skewed velocity profile (u_{max}/u_{av}=1.17) with velocity peak at 35% of the duct height from the bottom wall. Fig.23a - 23b shows the velocity profile along the vertical and horizontal axis generated by such a grid.Fig.24 -25 shows that for equal suction from both top and bottom walls the oulet velocity profile was also hub-skewed. Such a velocity profile

will not be desirable to combustion chamber during cruise operation when more amount of air is to pass through the flame tube i.e a centre peaked velocity profile is desirable. Fig.26 - 27 shows that such asymmetric velocity profile can be made centre peaked by adapting one side bleed (in the present case only from top wall) and thus making it suitable for all engine operating conditions. Further it can be seen that divergence angle of 30^0 requires less suction to achieve a centre peaked velocity profile as compared to 45^0 owing to smaller pressure gradient. However Fig.27 shows severe distortion due to seperation on the bottom wall due to the absence of suction from that wall, which is undesirable. This can be overcome by very low bleed rate of the order of 1-1.5% from the bottom wall while maintaining a higher bleed from the top wall to make the hub-skewed profile centre-peaked (Ref.4). But this operation was not possible as the present set up enabled to have only equal bleed rates from top and bottom walls. Fig.29 shows that asymmetric velocity profile gives less effectiveness owing to a very thick boundary layer. From Fig.30 it can be inferred that one side bleed lowers the effectiveness drastically owing to the fact that the flow is completely seperated at the wall where no bleed is done. This aspect is clearly seen in Fig.31 where bottom wall (no suction employed) gives a very low effectiveness.

Attempts were made to study the nature of the vortex formed with the help of flow visualisation techniques. Initially kerosene vapour was tried and when it failed to reveal the nature of the flow, flow visualisation with Titanium-tetra-chloride was attempted. Even though the smoke formed were denser this also got diffused near the vortex chamber. This is because

of very high intensities of turbulence in the vortex chamber.

PERFORMANCE WITH VORTEX GENERATORS

IMPOSING SYMMETRICAL INLET VELOCITY PROFILE

Initially experiments were carried out with vortex generators placed at a distance 0.4h (h is the height of the primary duct) upstream of the secondary inlet. Fig.33 shows that the vortex generators give a very small improvement in effectiveness for area ratio 2 and divergence angle 30^{O} at zero bleed rate. At other bleed rates vortex generators decrease the performance of the diffuser. This could be because the strength of the vortices shed from the vortex generators is not sufficient enough to overcome the sharp adverse pressure gradient and hence gives only a very slight improvement at zero bleed rate. At finite bleed rates the suction has a negative effect on the vortices shed and hence a large drop in effectiveness.

Later on experiments were carried out with vortex generators shifted to double the original distance upstream of the secondary inlet. Fig.34 shows that moving the vortex generators further upstream has not helped in improving the performance of the diffuser. Fig.35 shows that initial location of the vortex generators gives better performance. This result shows that by shifting the vortex generators further upstream of the secondary inlet the strength of the vortices shed decreases as it approaches the secondary inlet resulting in larger seperation losses.

CONCLUSIONS

1. Fence subtended angle of 25^{0} for 30^{0} divergent angle diffuser and fence subtended angle of 30^{0} for 45^{0} divergent angle diffuser gave the best effectiveness.
2. The non-uniformity of the inlet velocity profile decreases the effectiveness of the diffuser for all diffuser configurations.
3. Higher area ratio and higher divergence angle decreases the effectiveness of the hybrid diffuser.
4. Vortex chamber depression increases with bleed rate for all diffuser configurations.
5. The tail pipe improves the effectiveness in all cases.
6. The asymmetrically distorted inlet velocity profile gives a lower performance for all diffuser configurations.
7. The diffuser exit velocity profile is similiar but more distorted as compared to inlet velocity profile when equal amounts of suction is employed from top and bottom walls.
8. Differntial bleed can be used to improve the exit velocity profile.
9. Vortex generators improved the effectiveness to a small extent for zero bleed rate.

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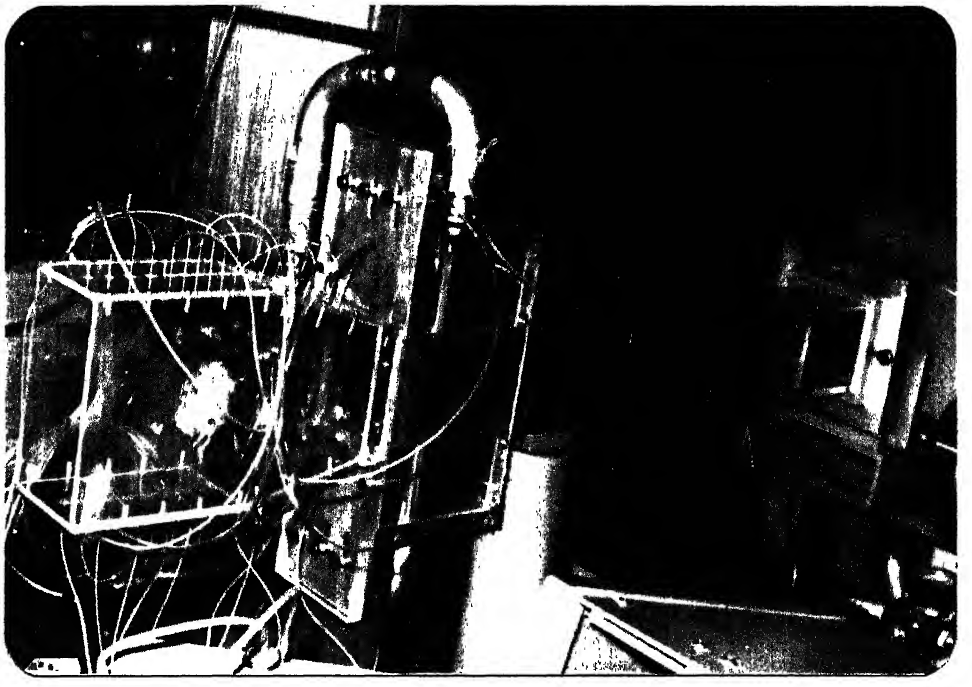
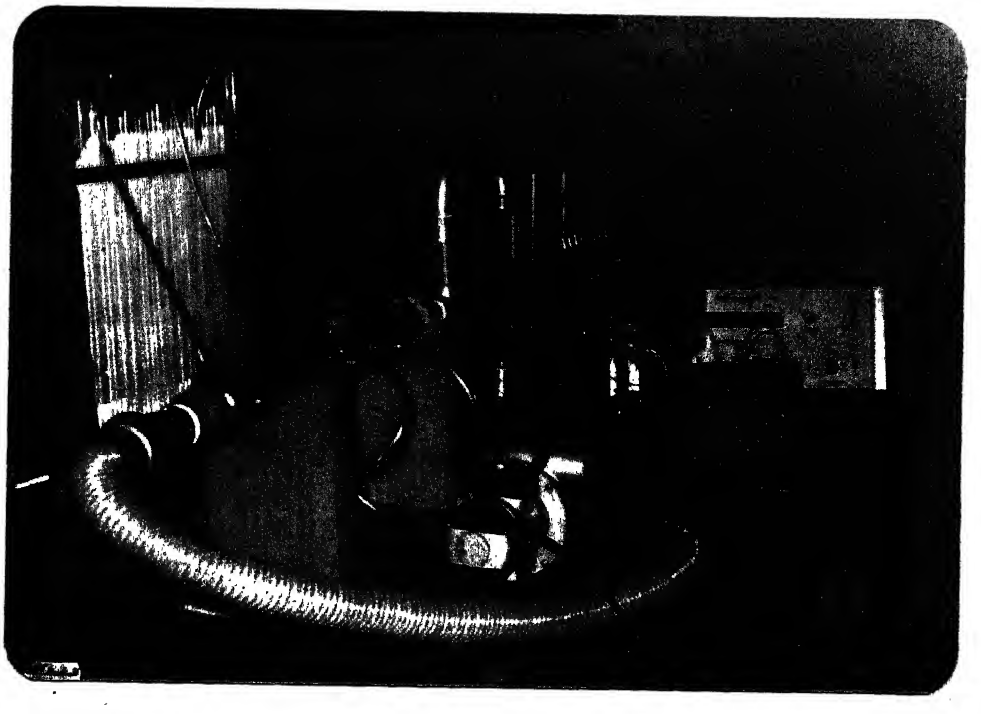
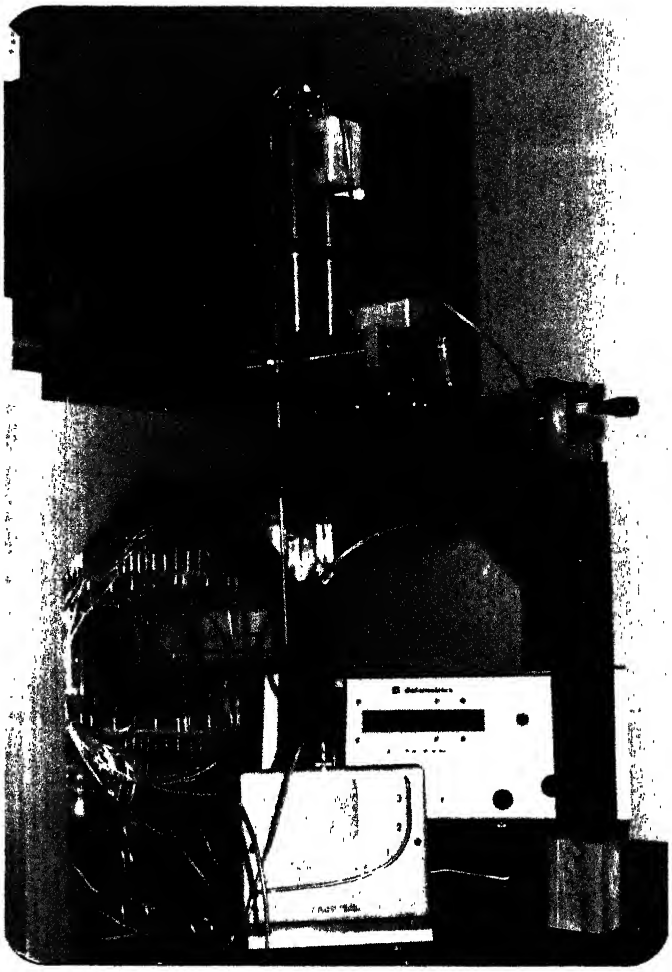


Photograph 1. Set up showing primary duct, vortex chamber and hybrid diffuser.



Photograph 2. Suction System.



Photograph 3. Traversing mechanism used for velocity and turbulence measurements.

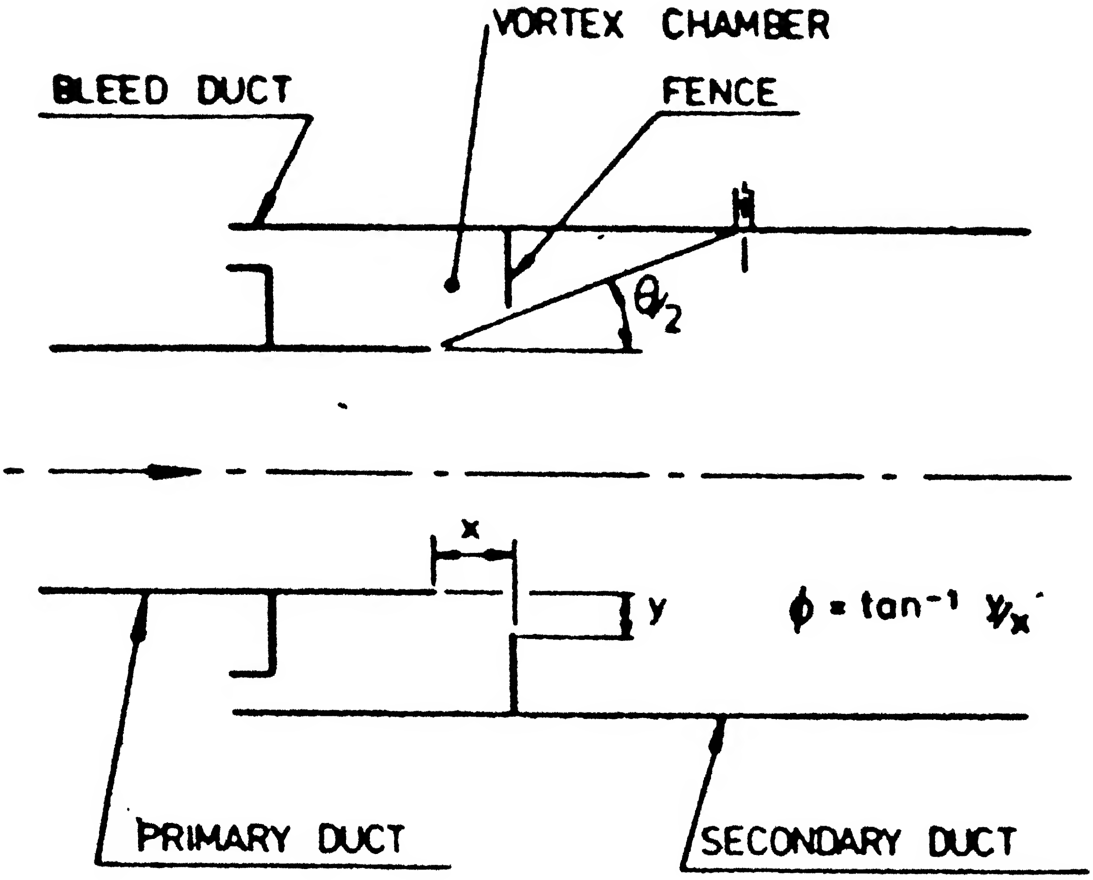


Fig. 1. *Sketch of Vortex Controlled diffuser*

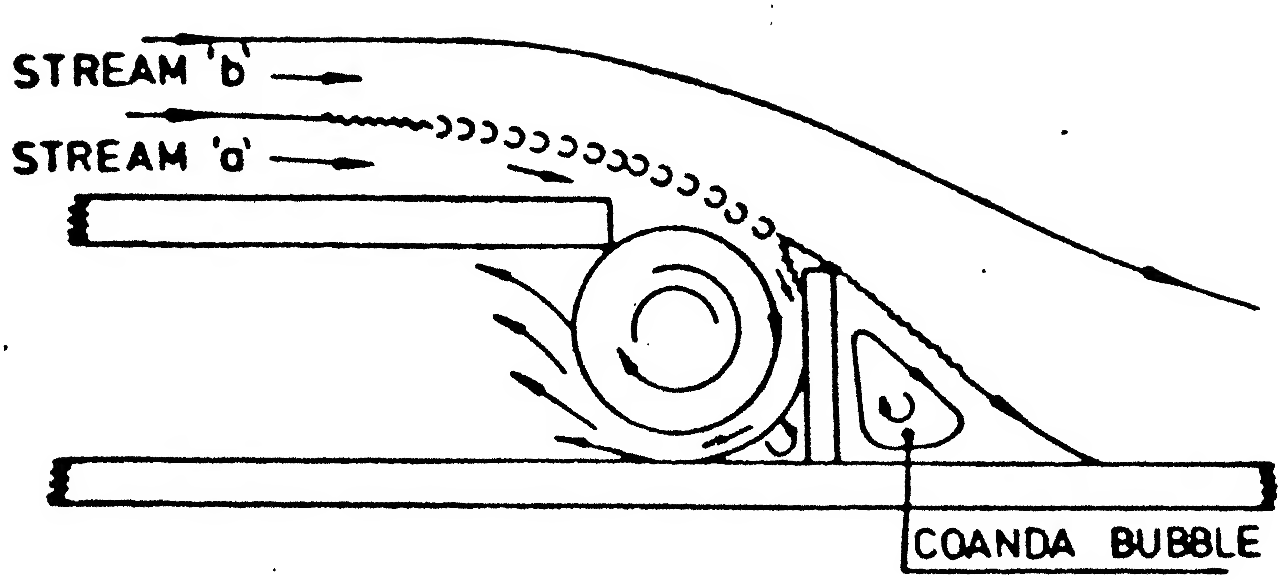


Fig. 2 Flow mechanism of vortex control

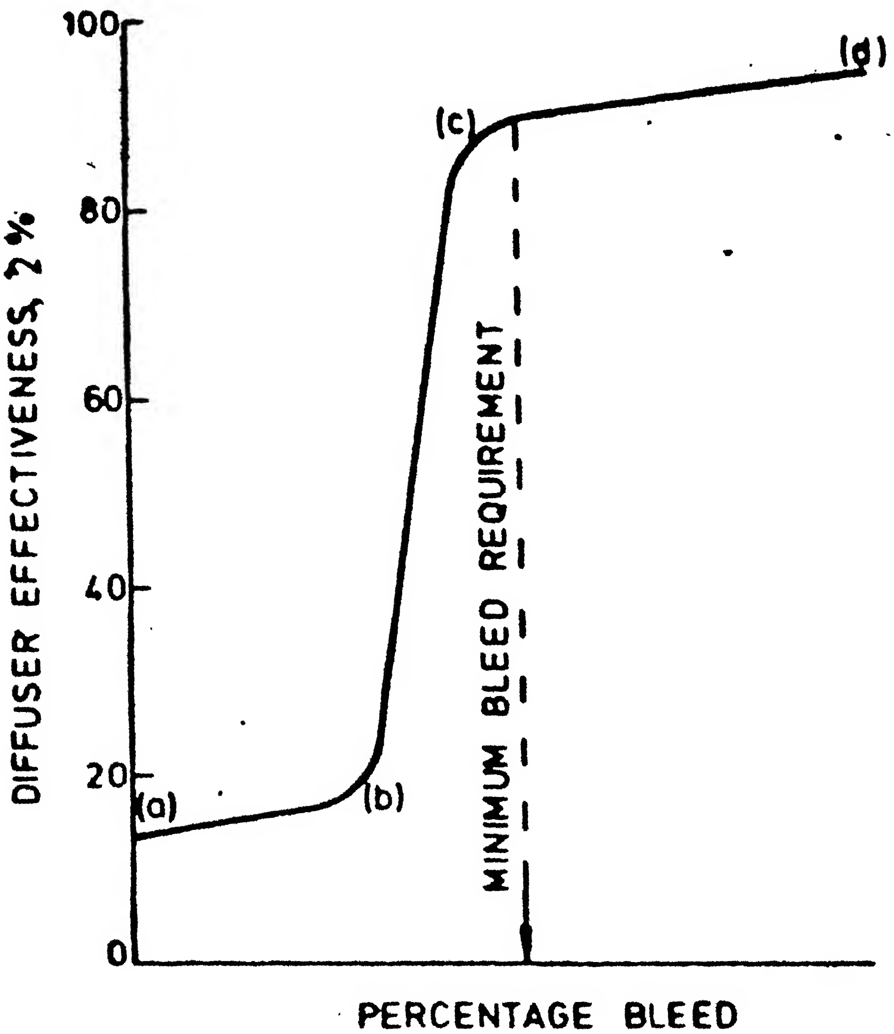


Fig. 3 Typical diffuser characteristic

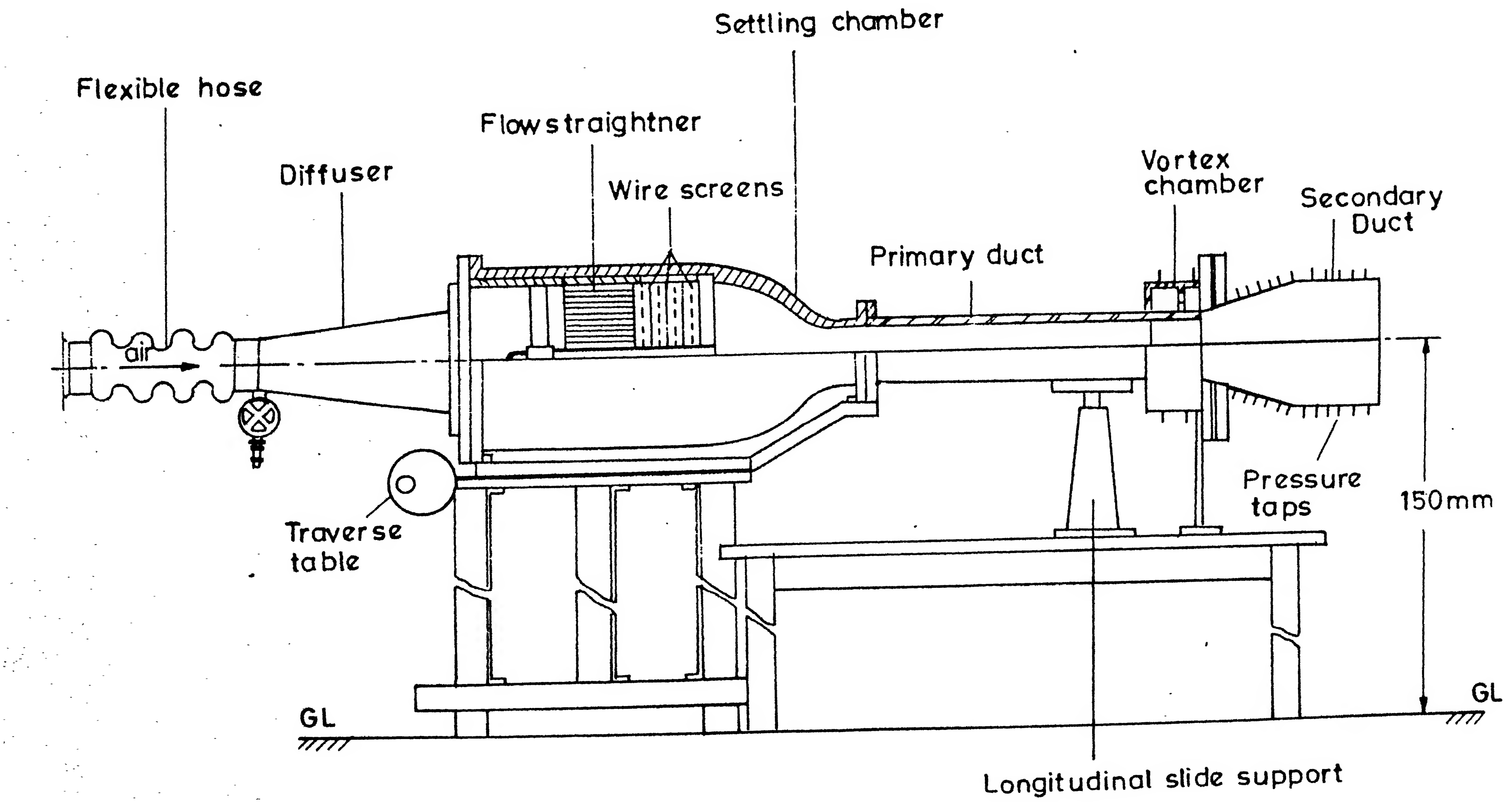


FIG.4 LAYOUT OF THE TUNNEL

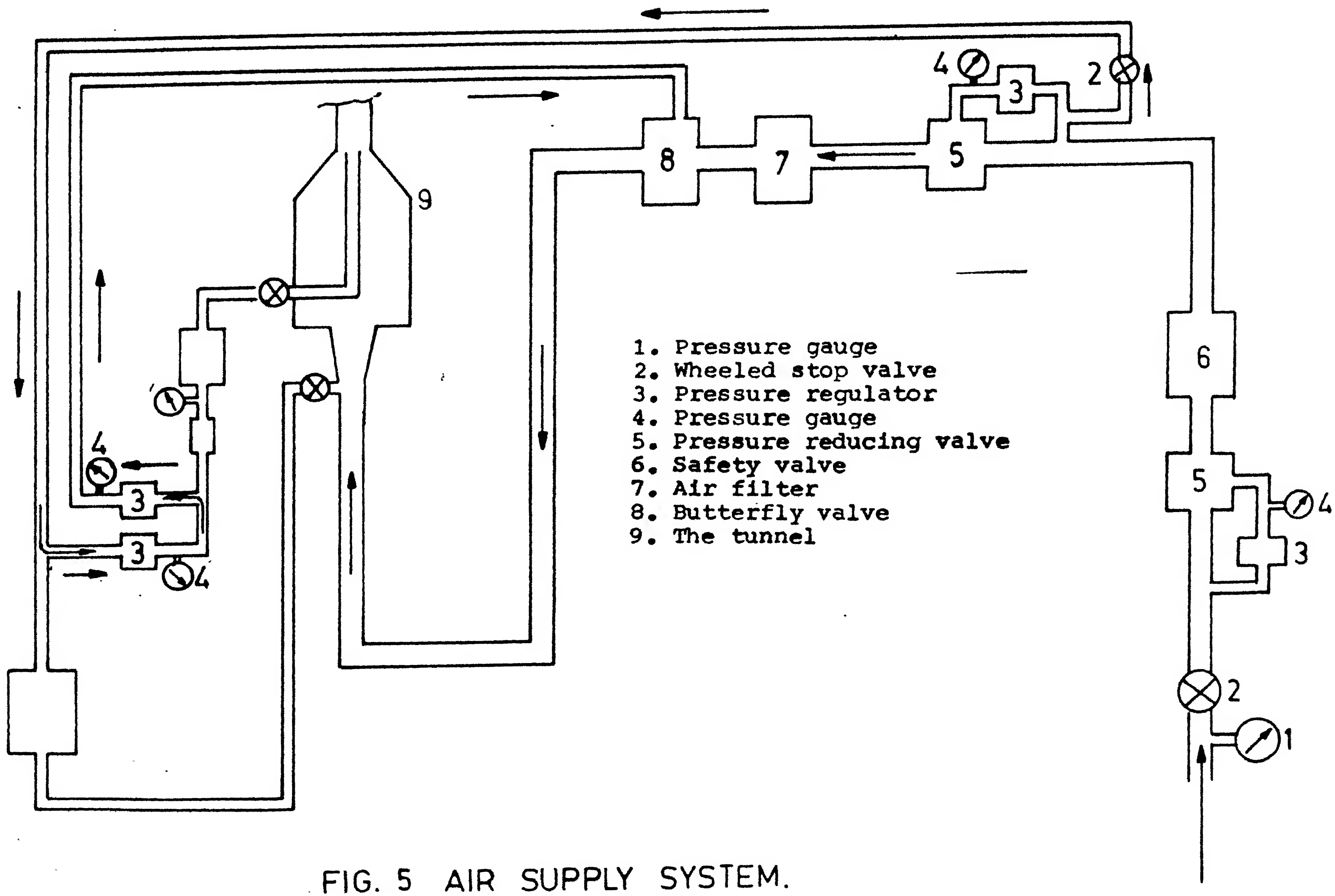


FIG. 5 AIR SUPPLY SYSTEM.

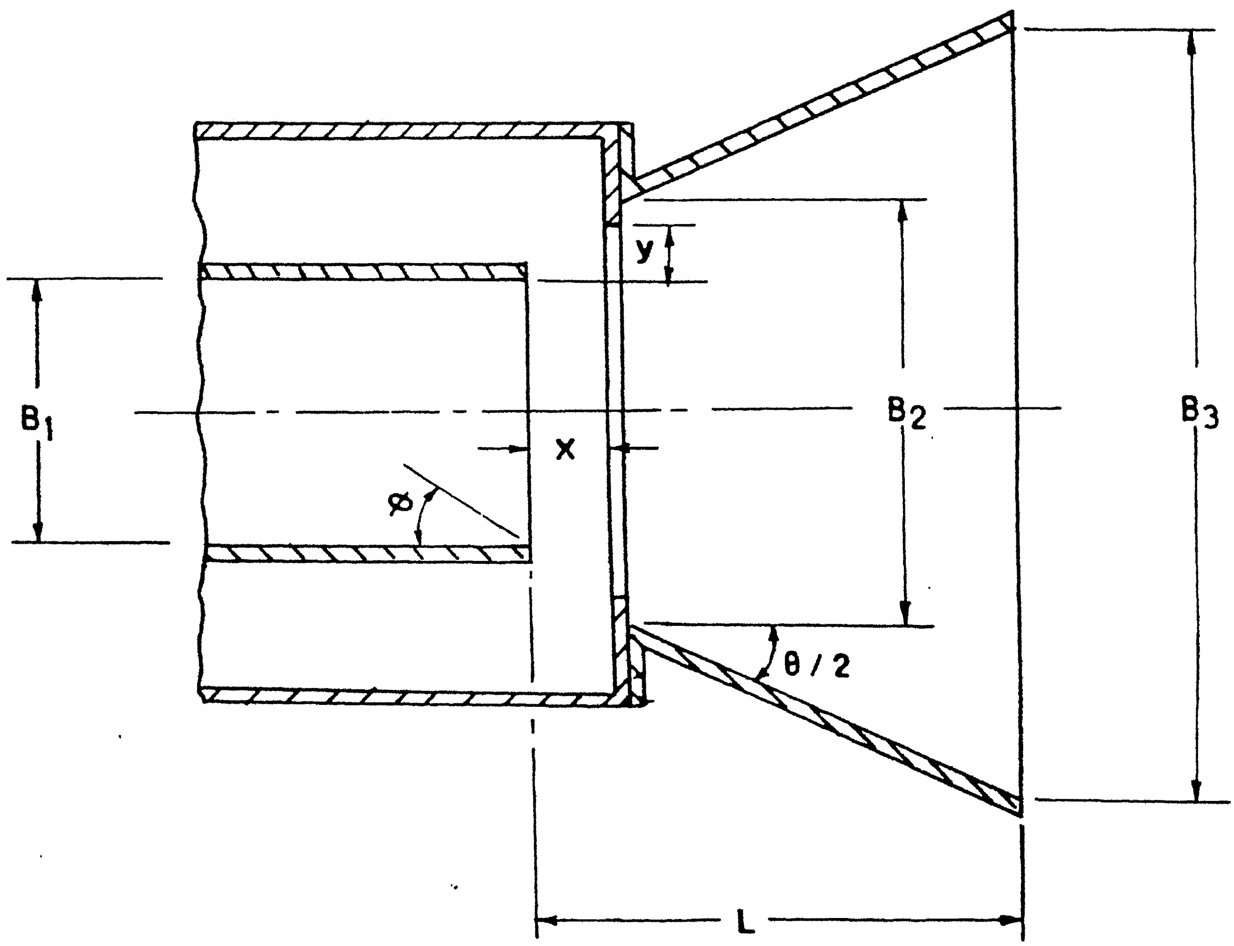


FIG. 6 GEOMETRIC PARAMETERS

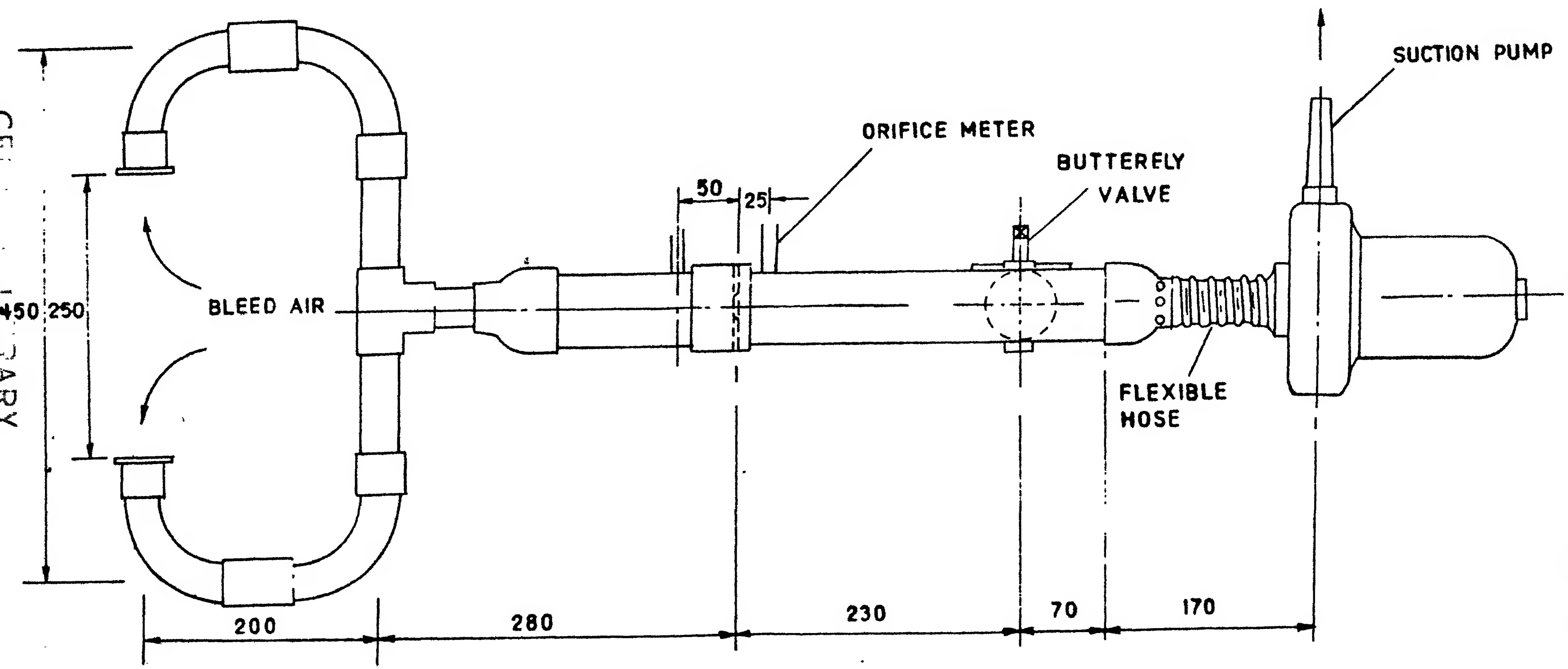


FIG.7 SUCTION SYSTEM

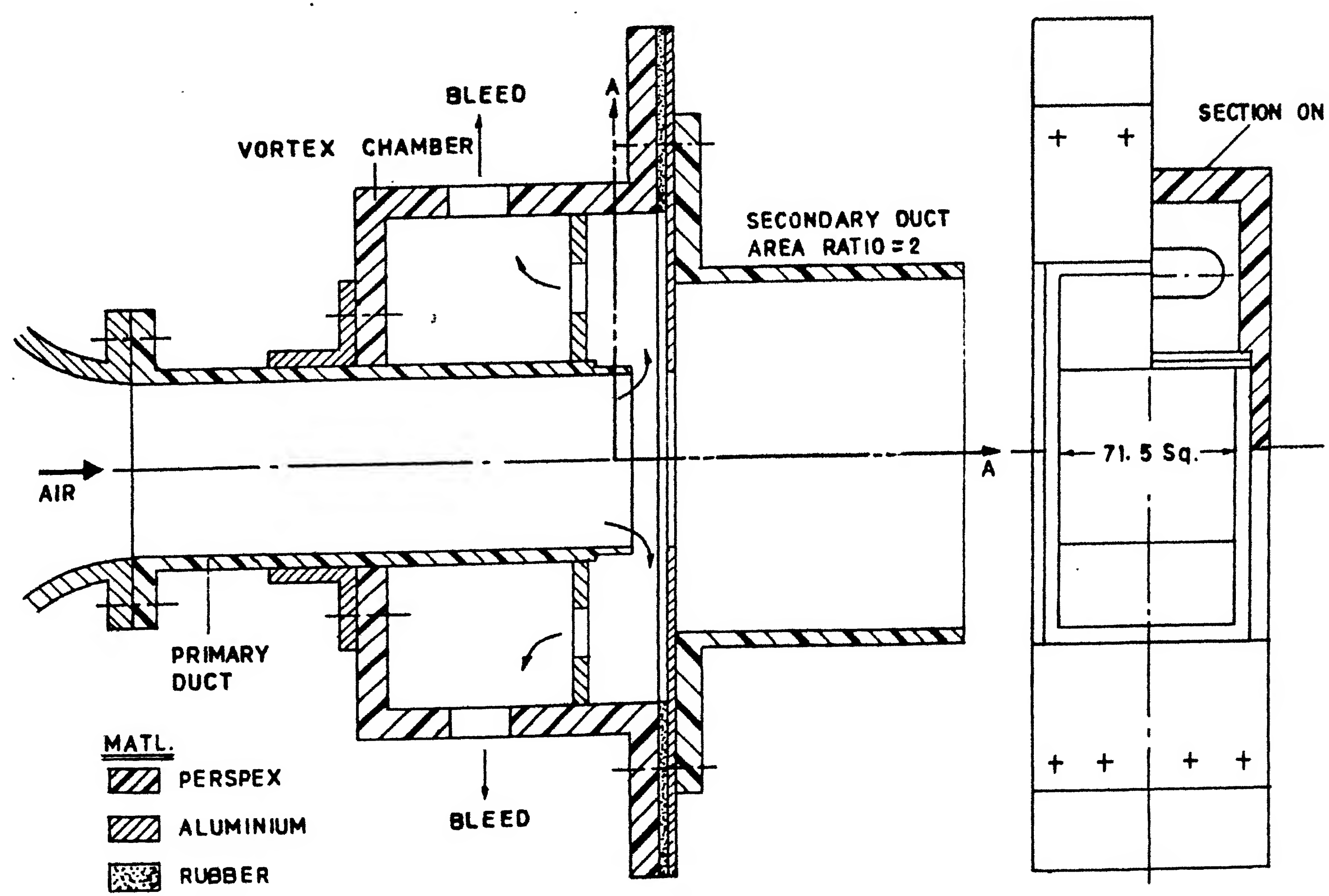


FIG.8 SCHEMATIC OF VORTEX FLOWCONTROLLED DIFFUSER

TUBE SPACINGS FOR $V_{max}/V_{av} = 1.10$

1st tube = 2.4 mm

2nd. " = 4.8 "

3rd " = 7.3 "

4th " = 9.9 "

5th " = 12.8 "

HYPODERMIC TUBE EXTERNAL DIA. = 1mm

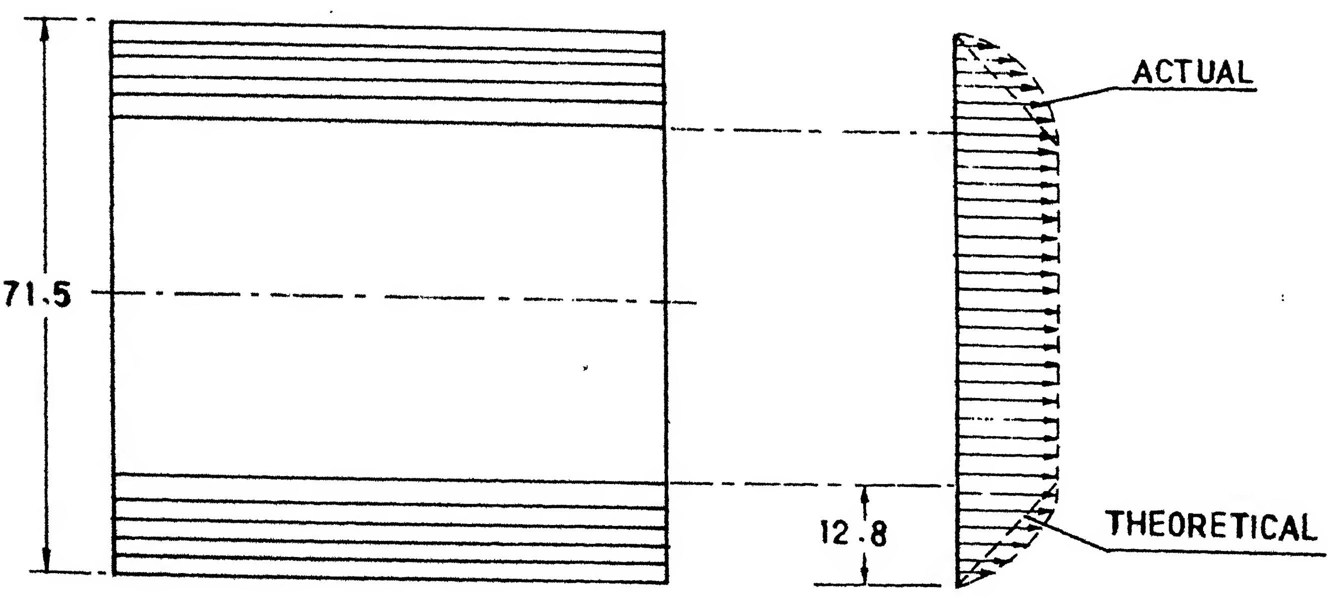
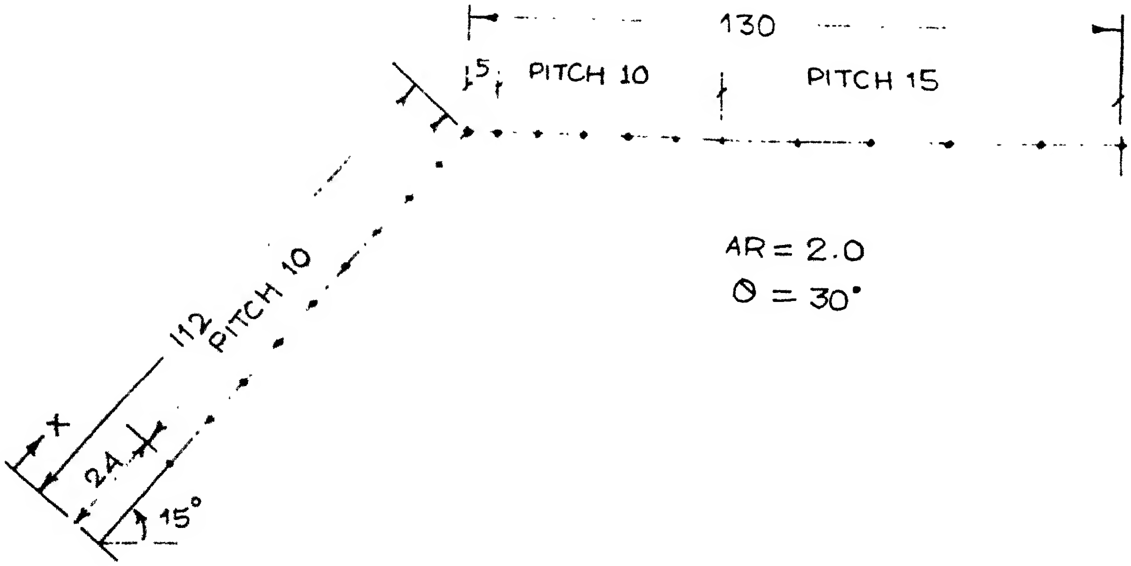


FIG. 9 GAUZE SCREEN DESIGN



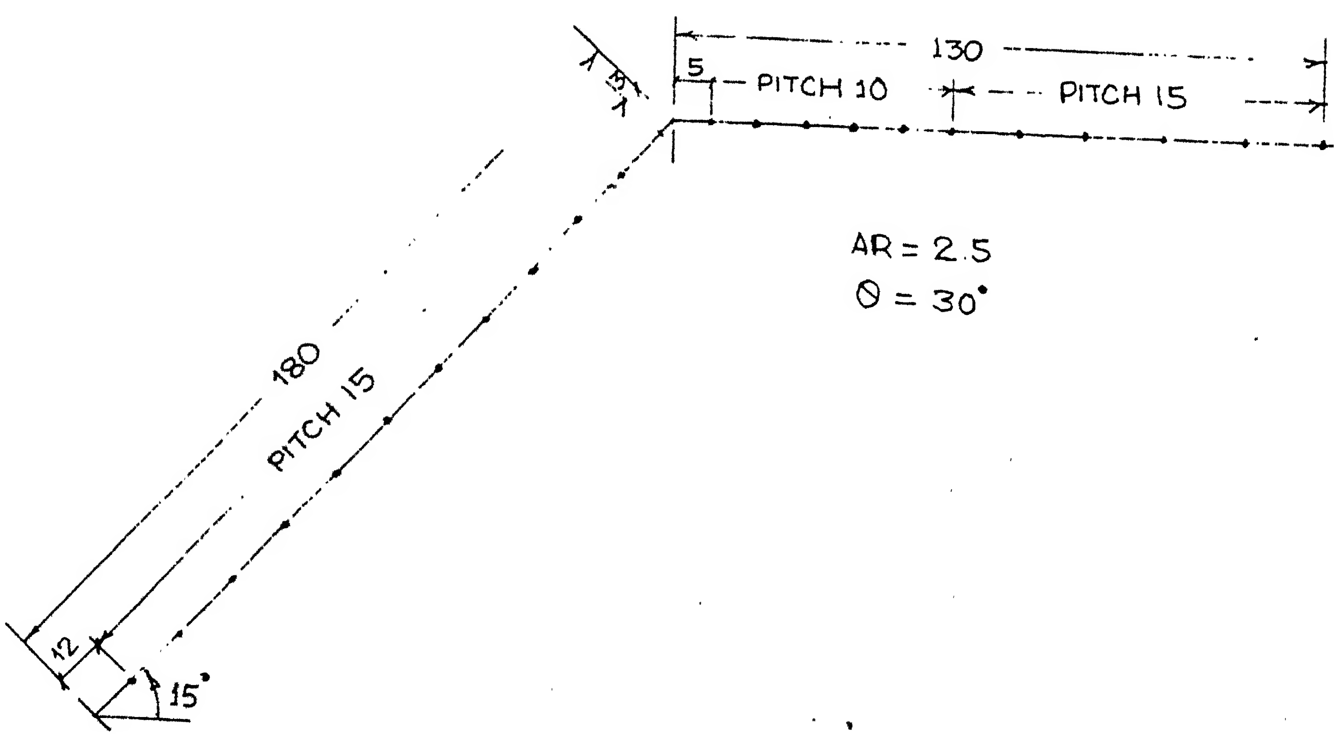


FIG 10 DETAILS OF TAPPING IN HYBRID DIFFUSERS

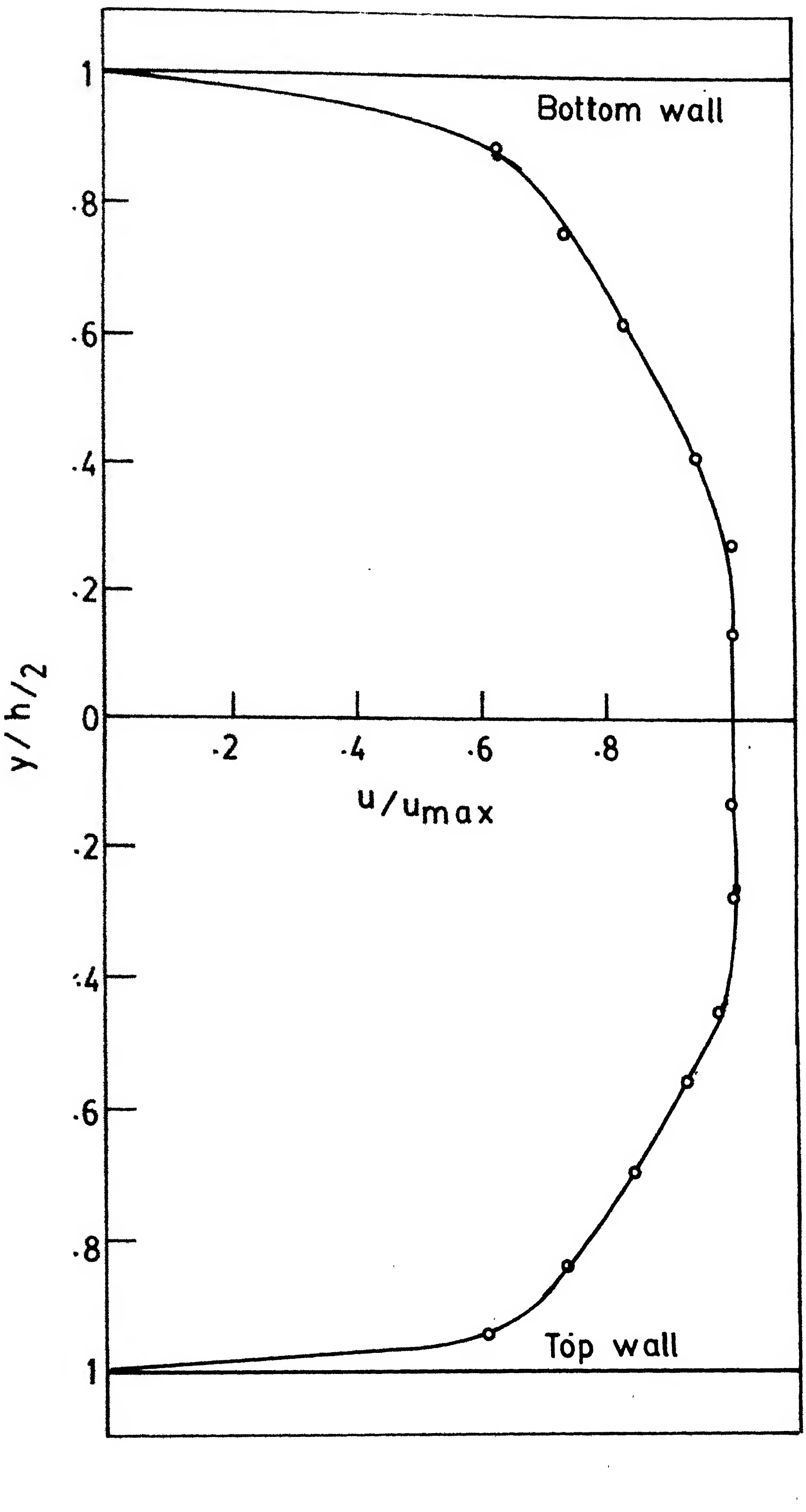


FIG. 11a VELOCITY PROFILE ALONG THE VERTICAL AXIS AT THE EXIT OF THE PRIMARY DUCT

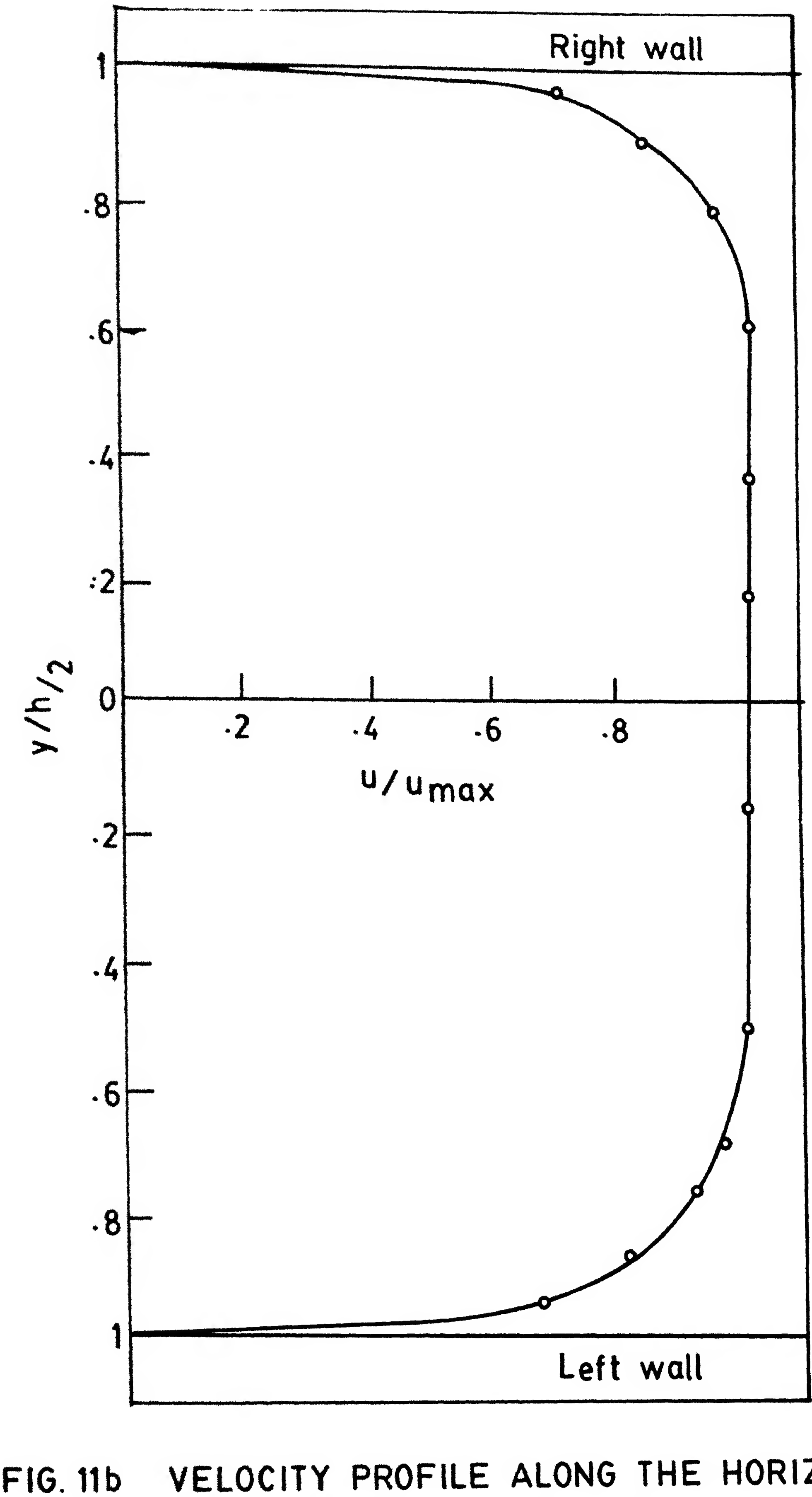


FIG. 11b VELOCITY PROFILE ALONG THE HORIZONTAL AXIS AT THE EXIT OF THE PRIMARY DUCT

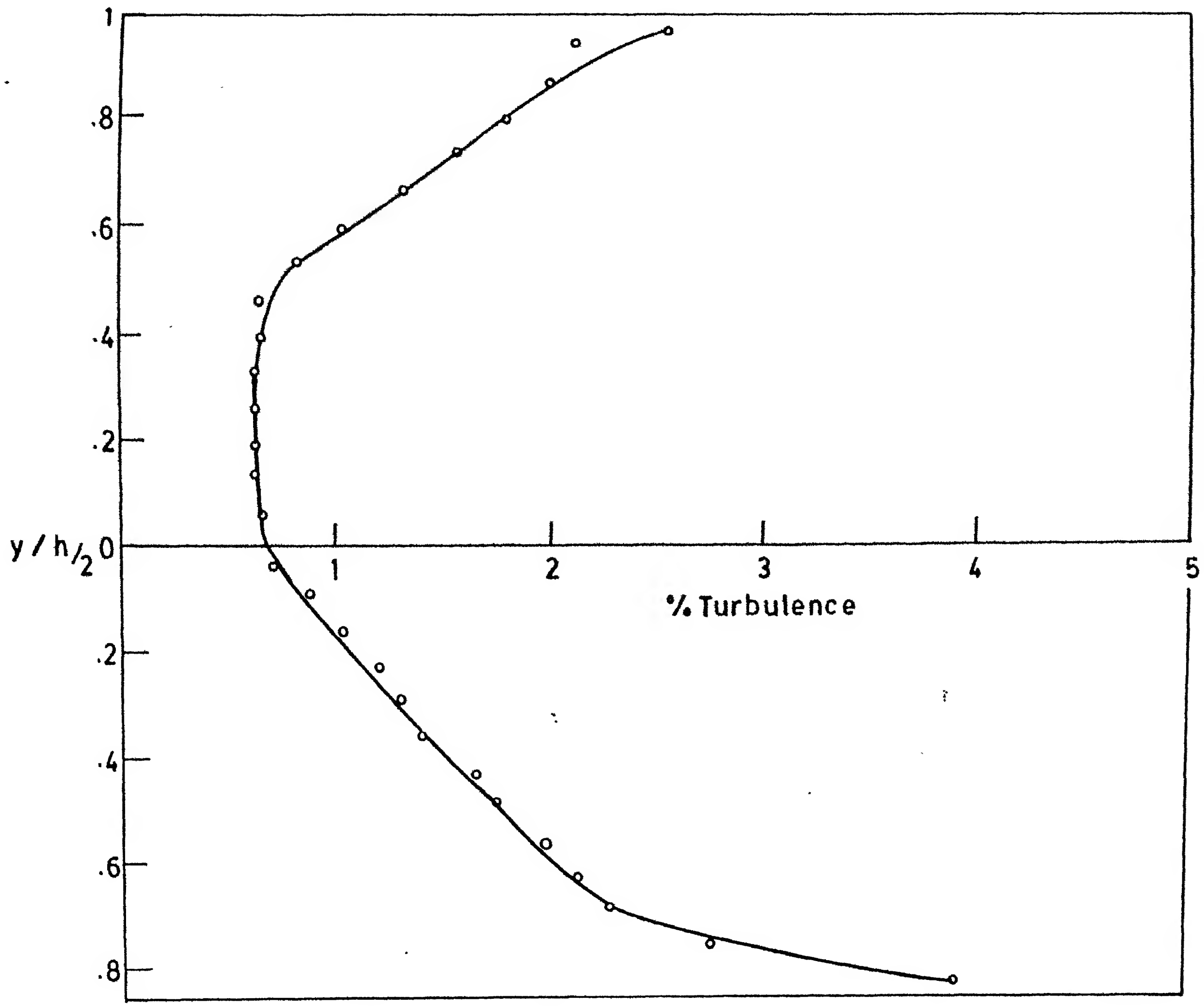


FIG. 12 STREAMWISE TURBULENCE INTENSITY DISTRIBUTION AT THE EXIT OF PRIMARY DUCT

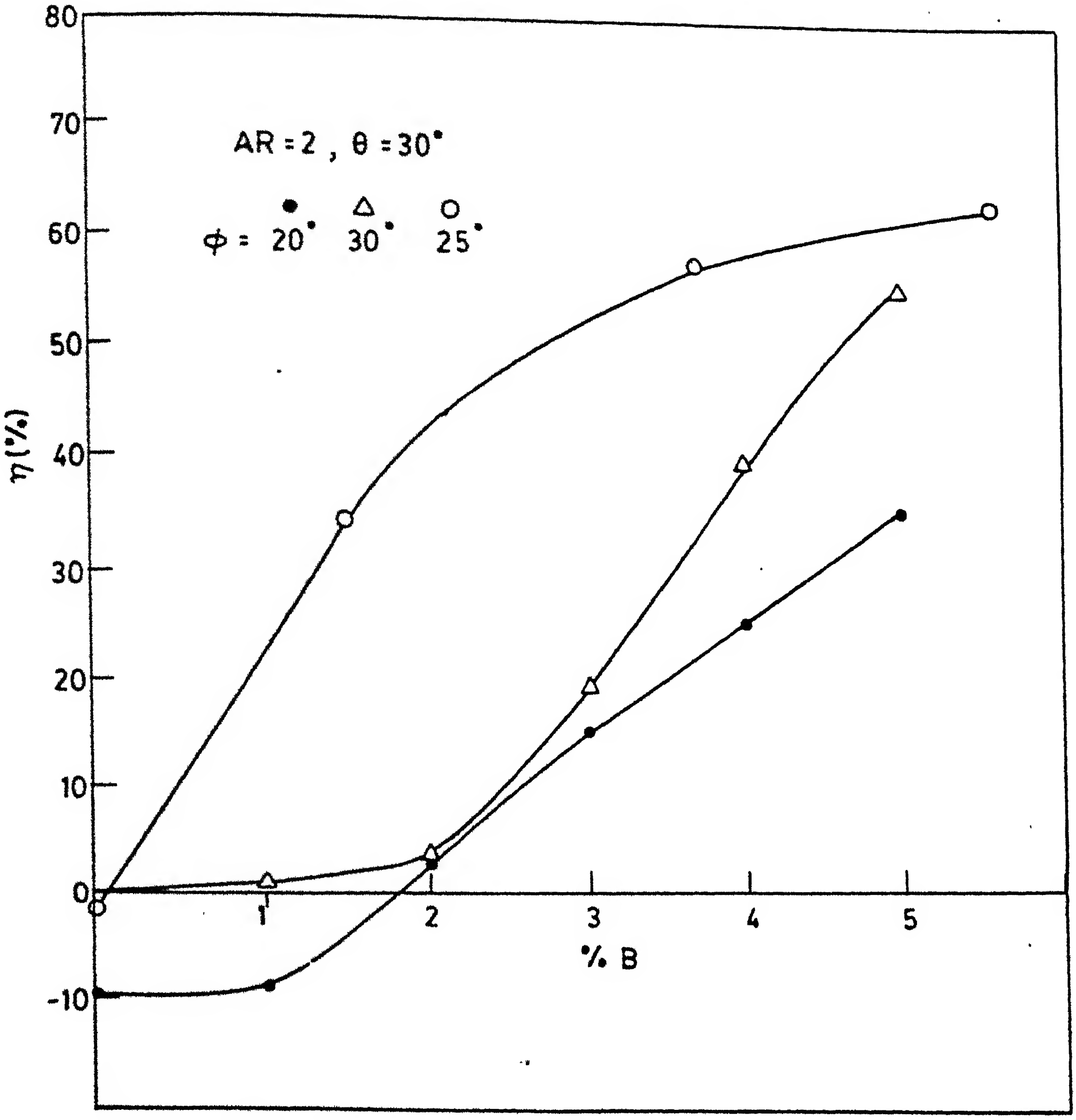


FIG. 13 EFFECT OF SUCTION ON DIFFUSER EFFECTIVENESS

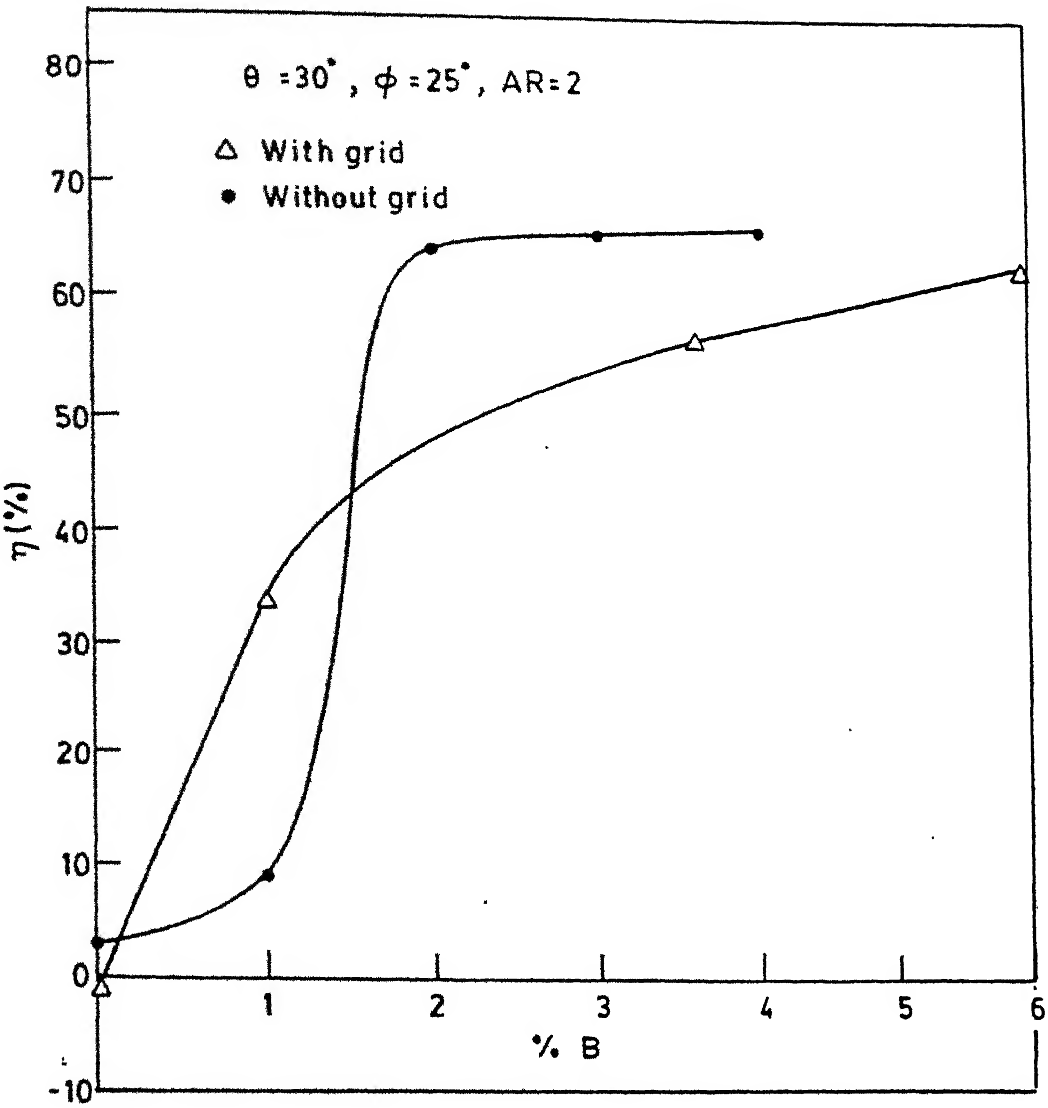


FIG. 14 COMPARISON OE DIFFUSER EFFECTIVENESS WITH UNIFORM AND NONUNIFORM INLET VELOCITY PROFILES

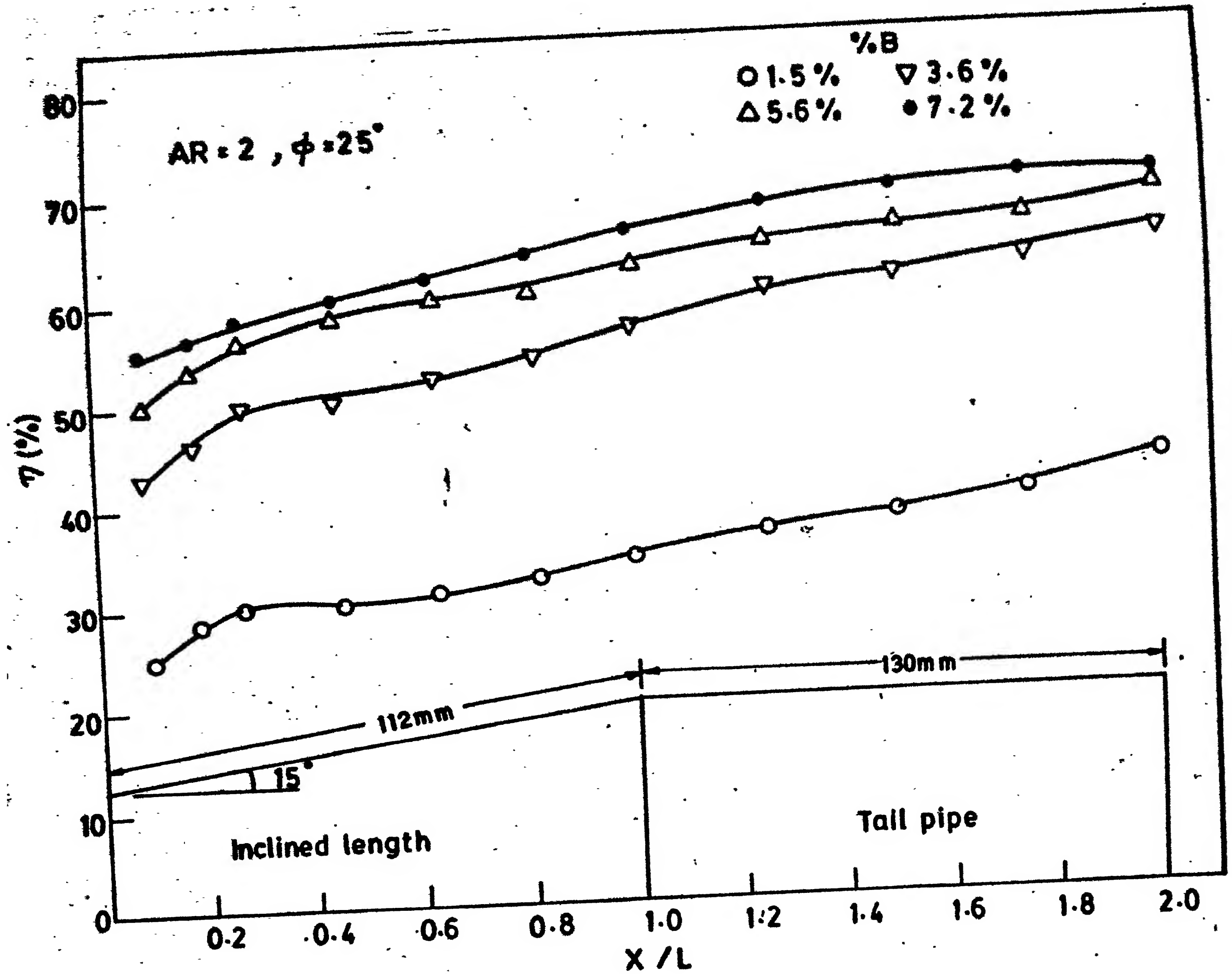


FIG. 15 DIFFUSER EFFECTIVENESS VARIATION ALONG THE LENGTH OF SECONDARY DUCT

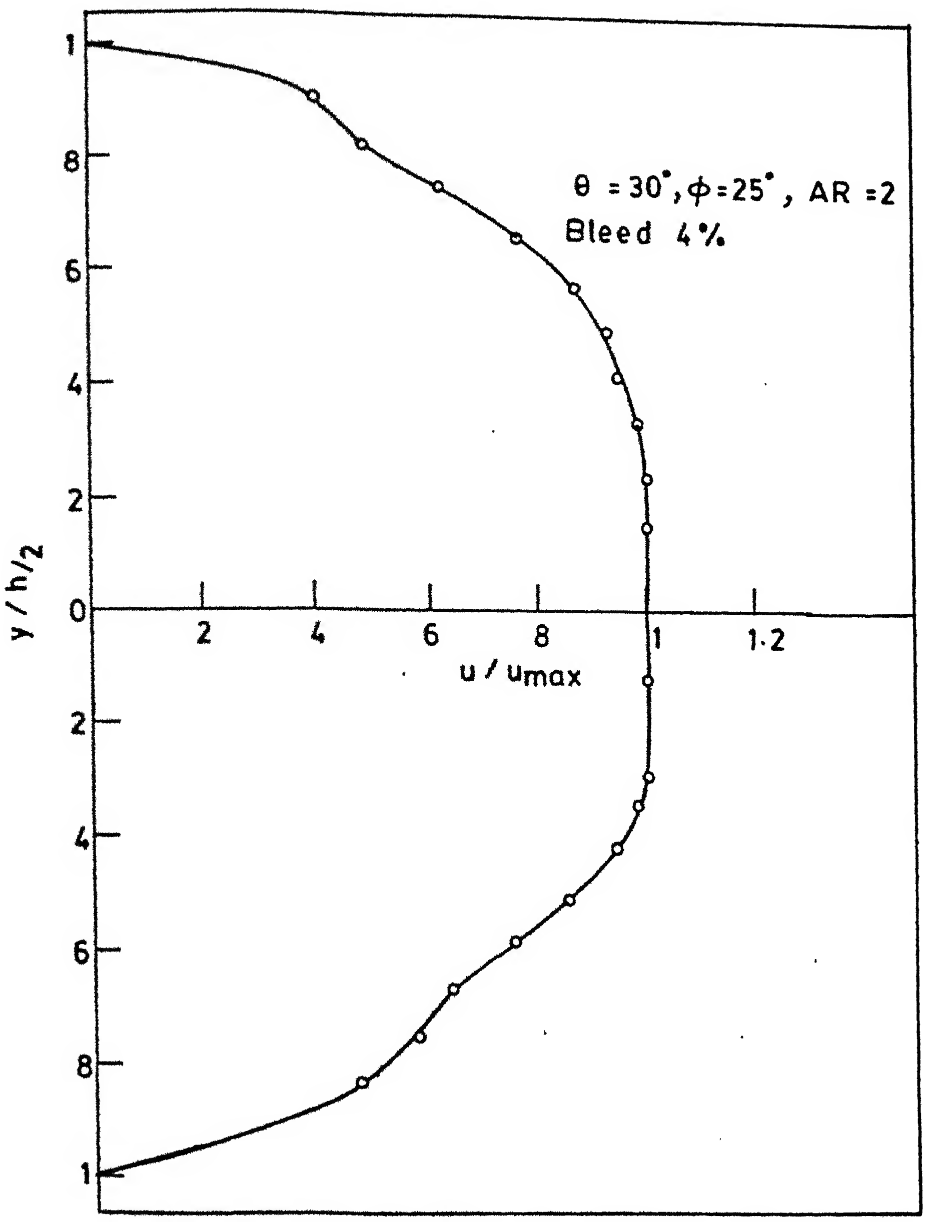


FIG. 16a DIFFUSER EXIT VELOCITY PROFILE ALONG THE HORIZONTAL AXIS

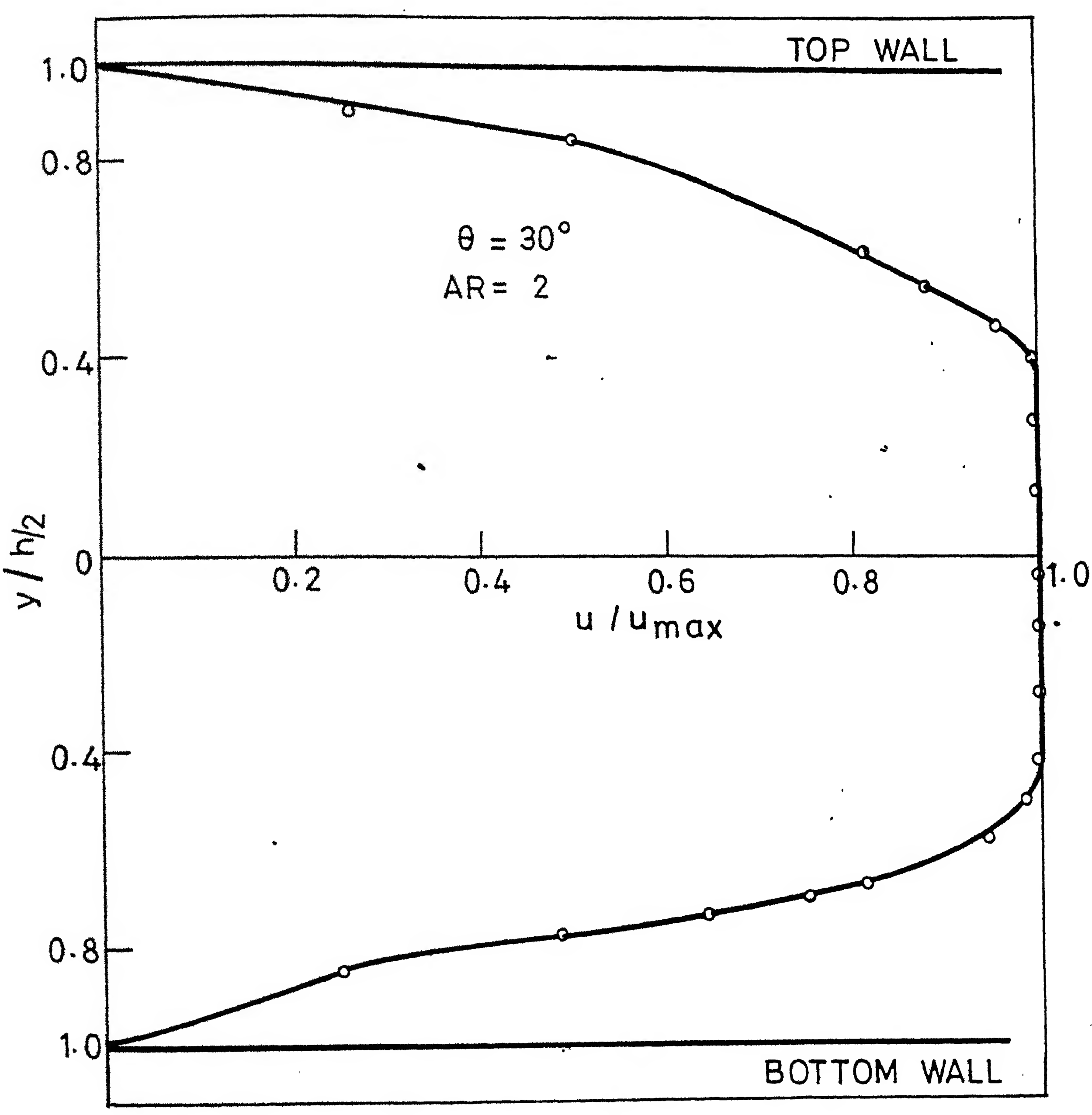


FIG. 16b VELOCITY TRAVERSE ALONG THE VERTICAL AXIS AT THE END OF HYBRID DIFFUSER

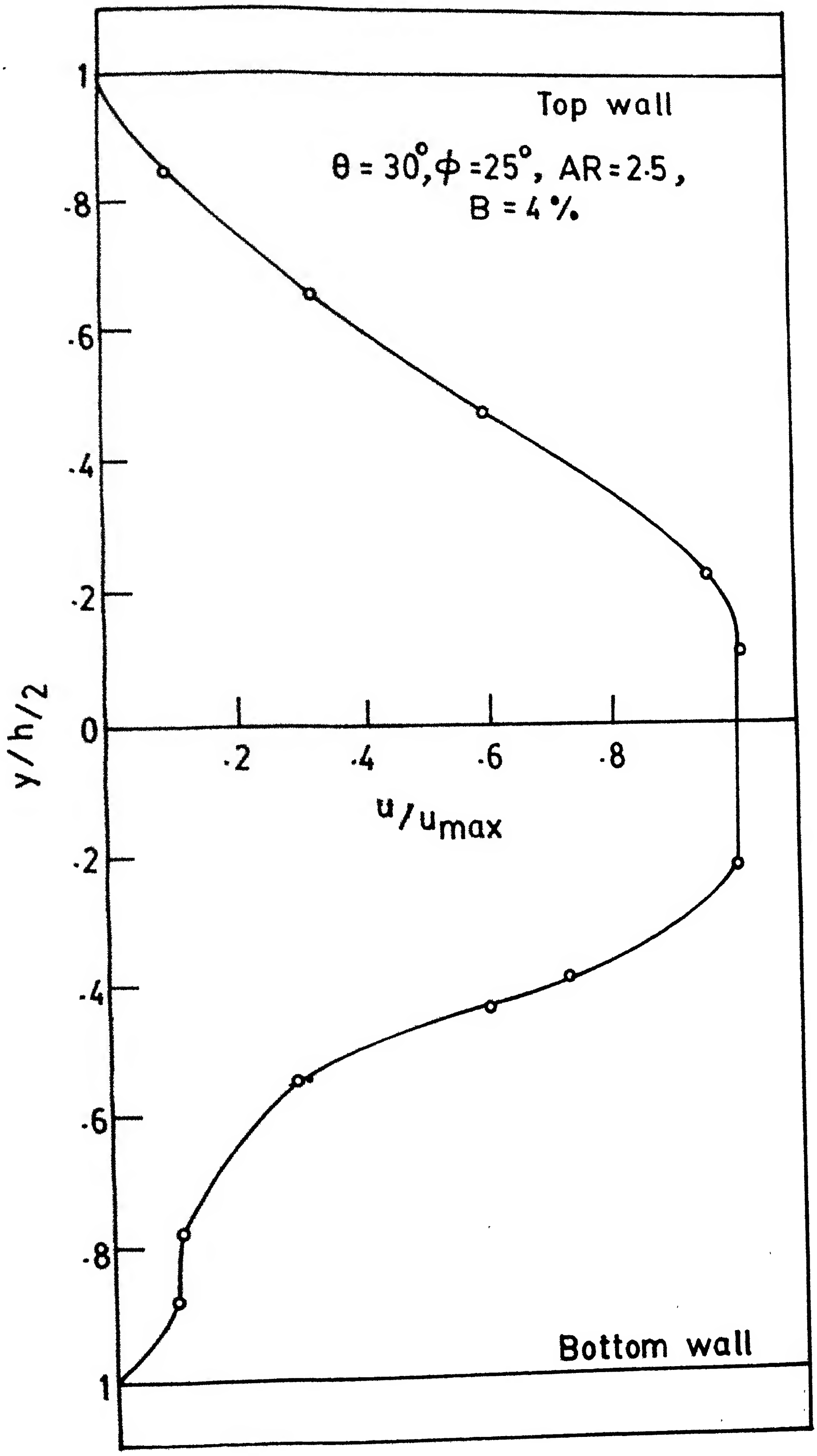


FIG. 17 a EXIT VELOCITY PROFILE ALONG THE VERTICAL AXIS

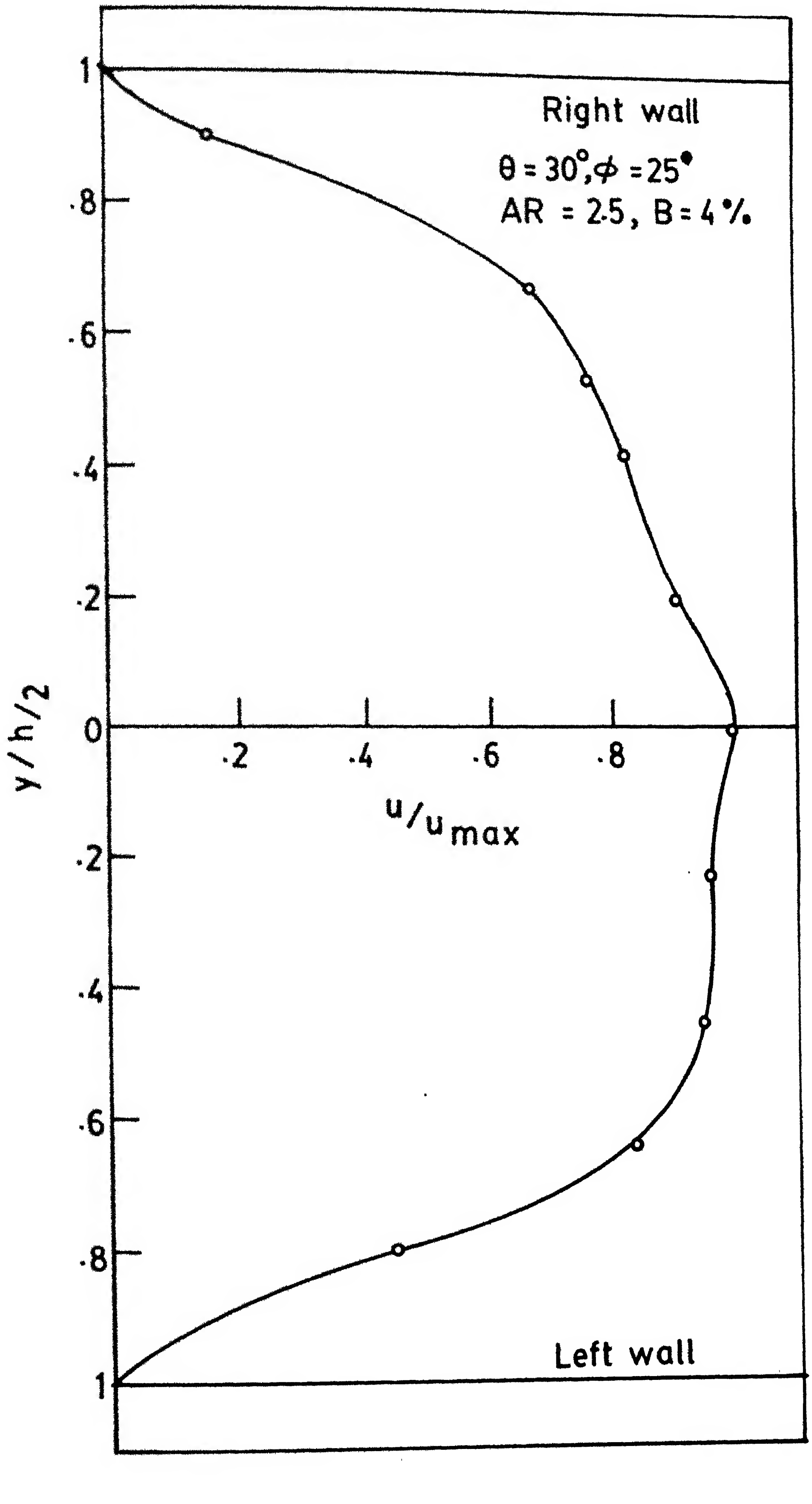


FIG. 17 b EXIT VELOCITY PROFILE ALONG THE HORIZONTAL AXIS.

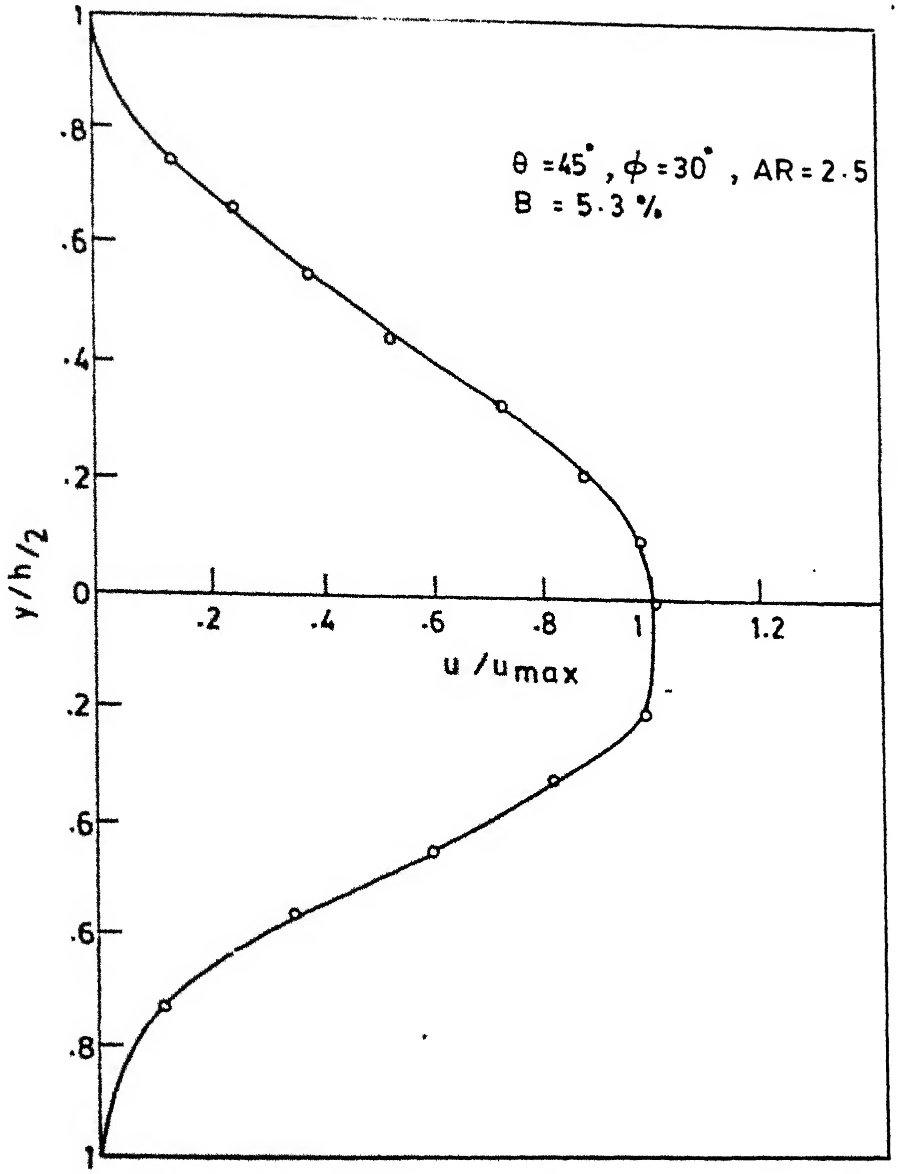


FIG 18 DIFFUSER EXIT VELOCITY PROFILE WITH NONUNIFORM INLET FLOW

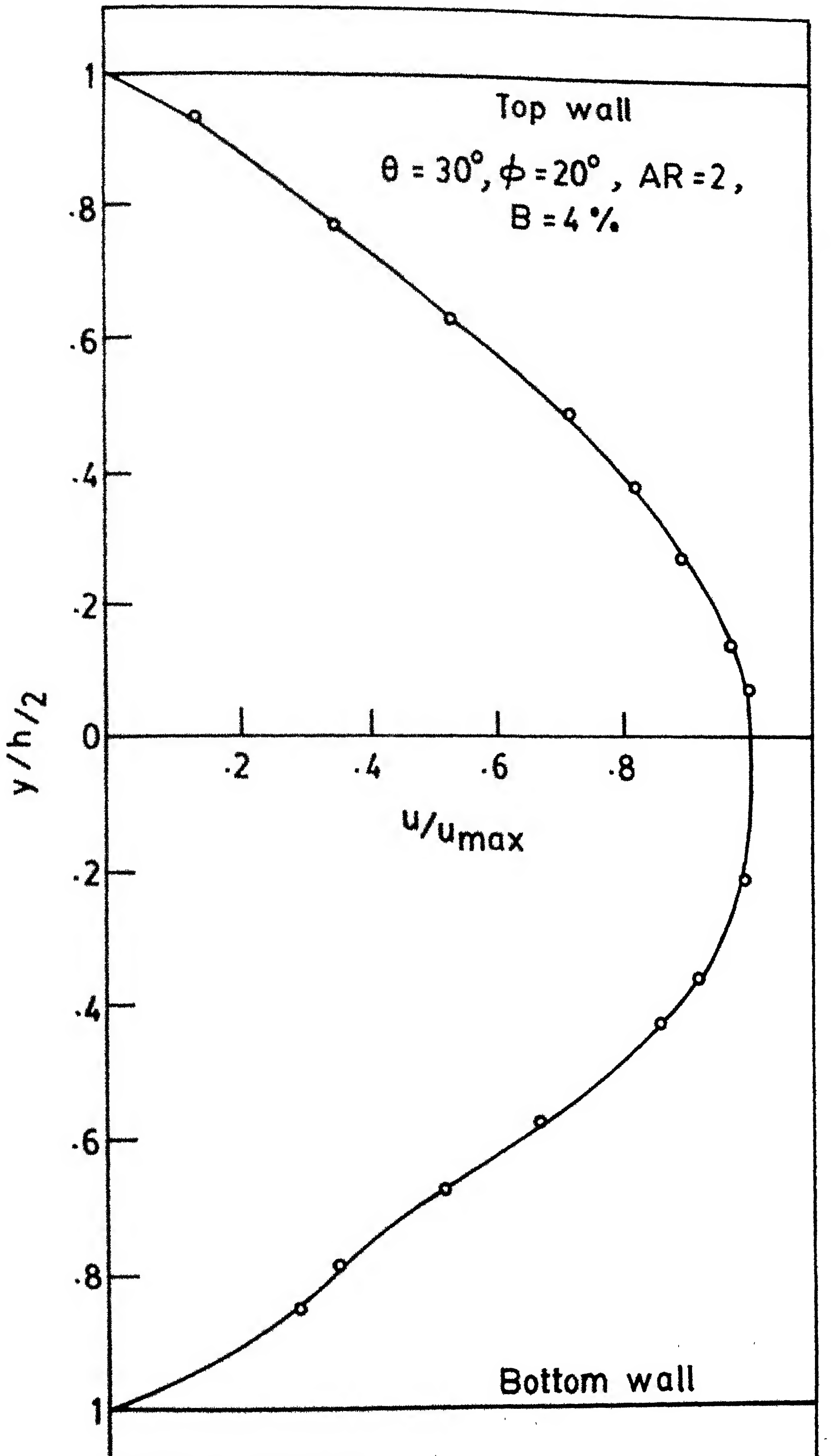


FIG. 19a EXIT VELOCITY PROFILE ALONG THE VERTICAL AXIS TAKEN AT THE CENTRE OF THE SECONDARY DUCT

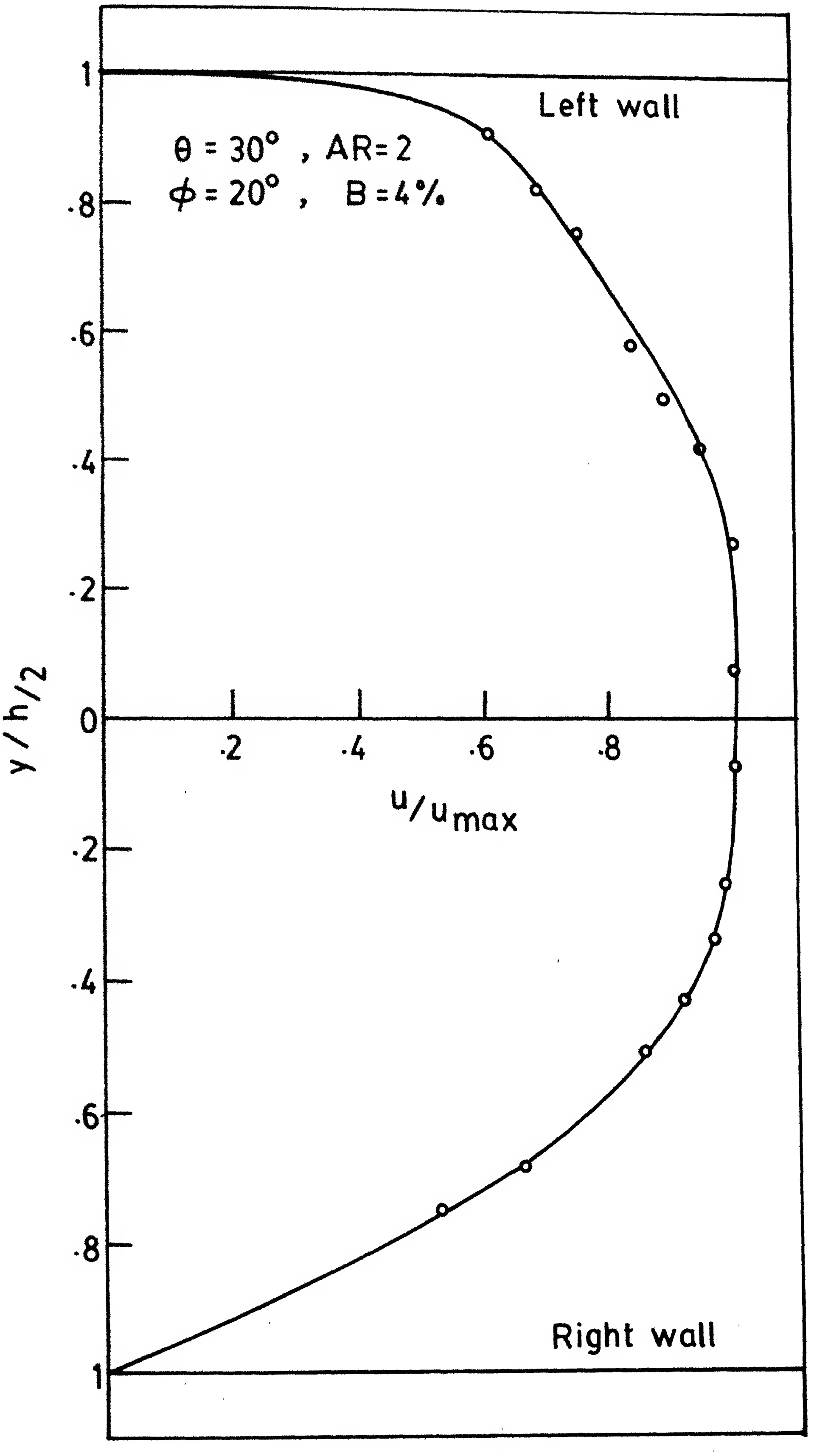


FIG. 19 b EXIT VELOCITY PROFILE ALONG THE HORIZONTAL AXIS TAKEN AT THE CENTRE OF THE SECONDAR

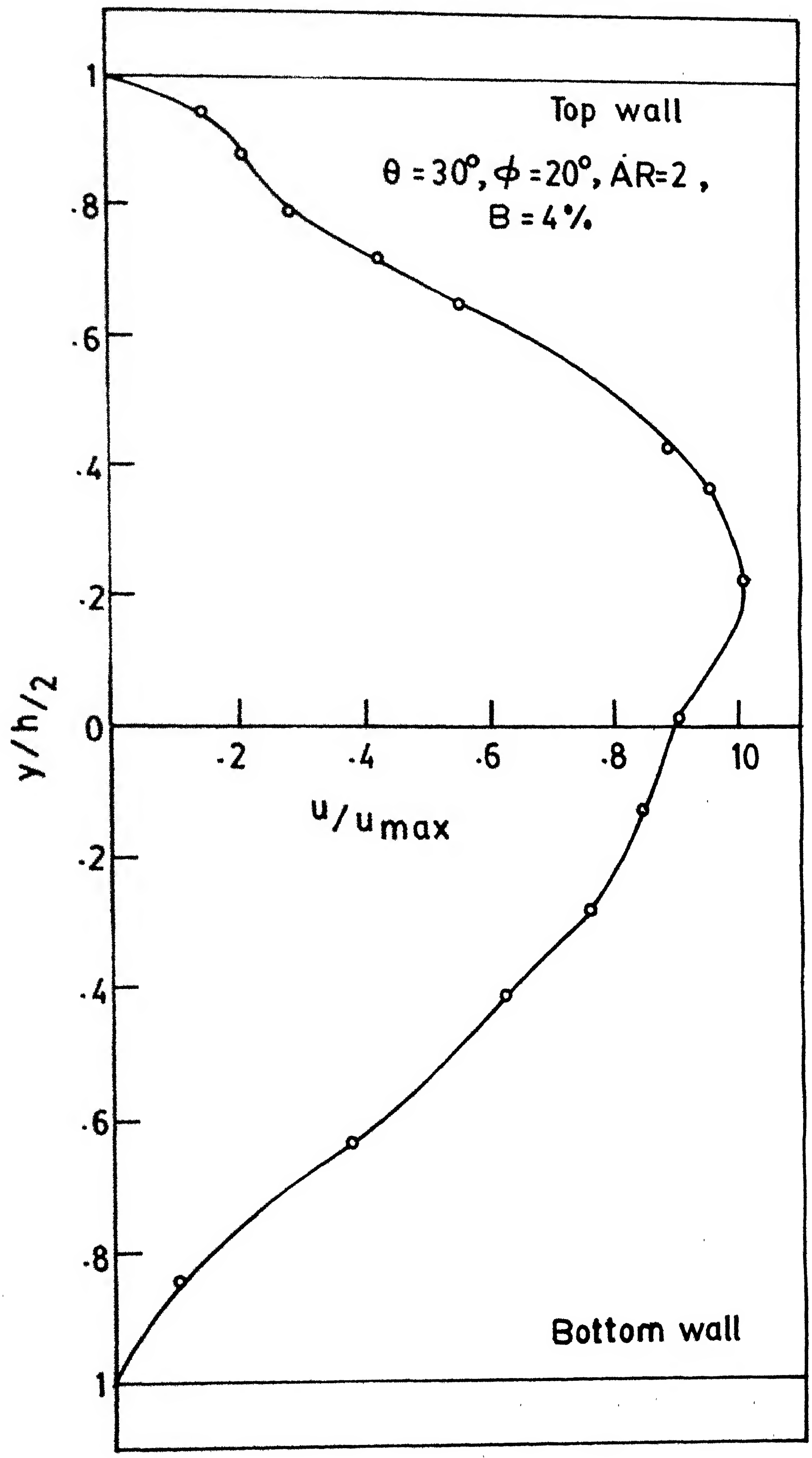


FIG. 19c EXIT VELOCITY PROFILE ALONG THE VERTICAL AXIS TAKEN AT A DISTANCE w/4 FROM THE RIGHT WALL

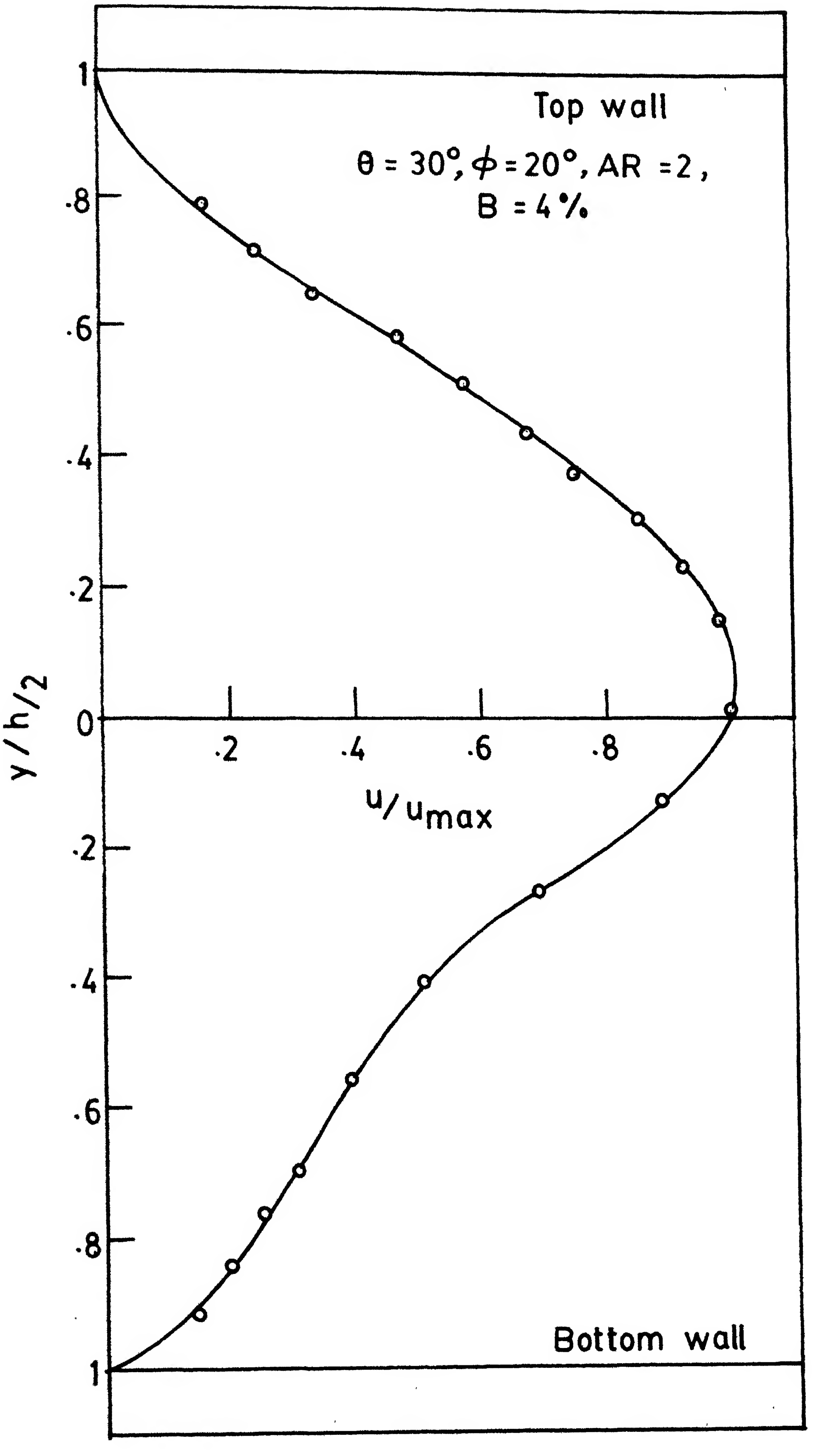


FIG. 19d EXIT VELOCITY PROFILE ALONG THE VERTICAL AXIS TAKEN AT A DISTANCE w/4 FROM THE LEFT WALL

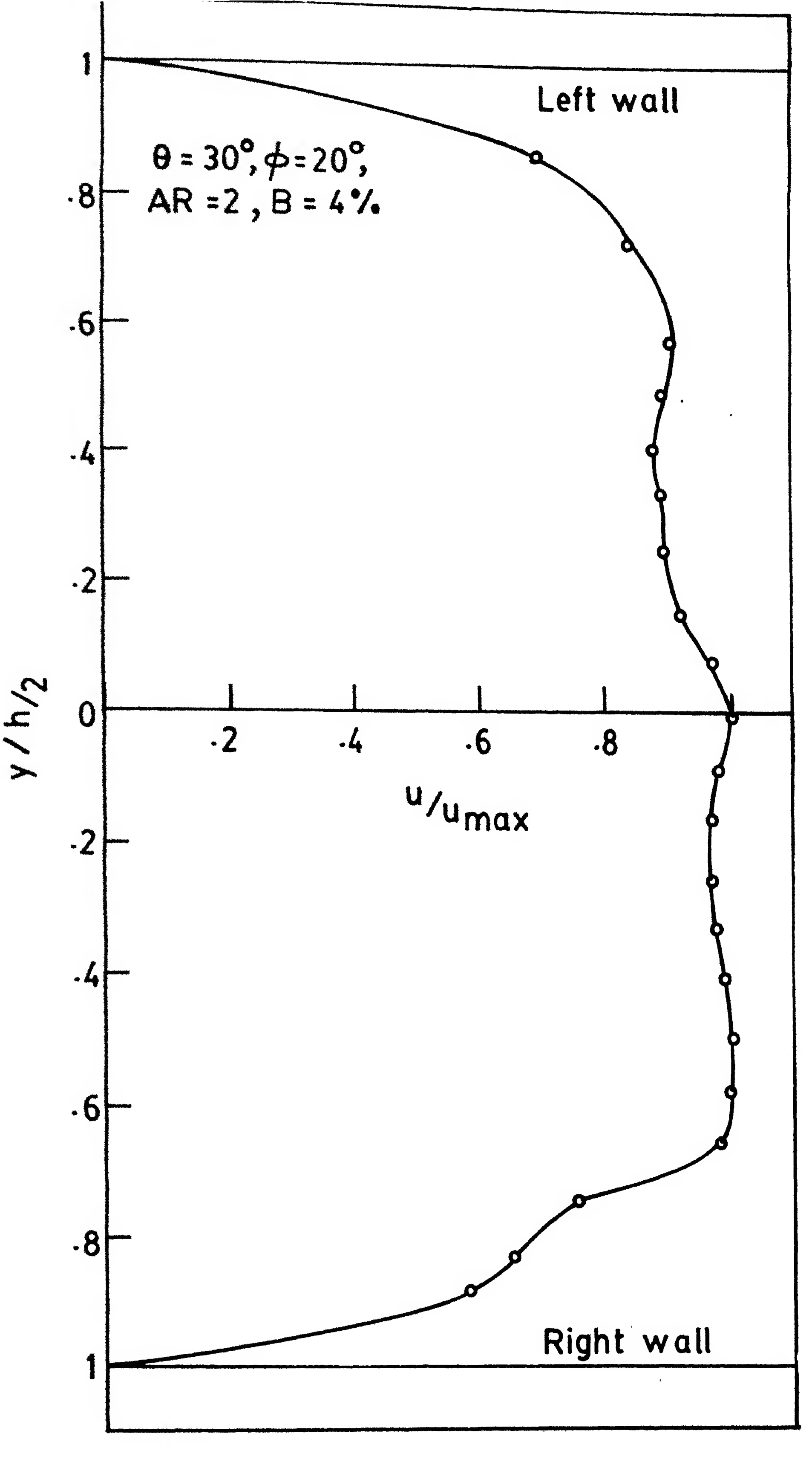


FIG. 19e EXIT VELOCITY PROFILE ALONG THE HORIZONTAL AXIS TAKEN AT A DISTANCE h/4 FROM THE TOP WALL

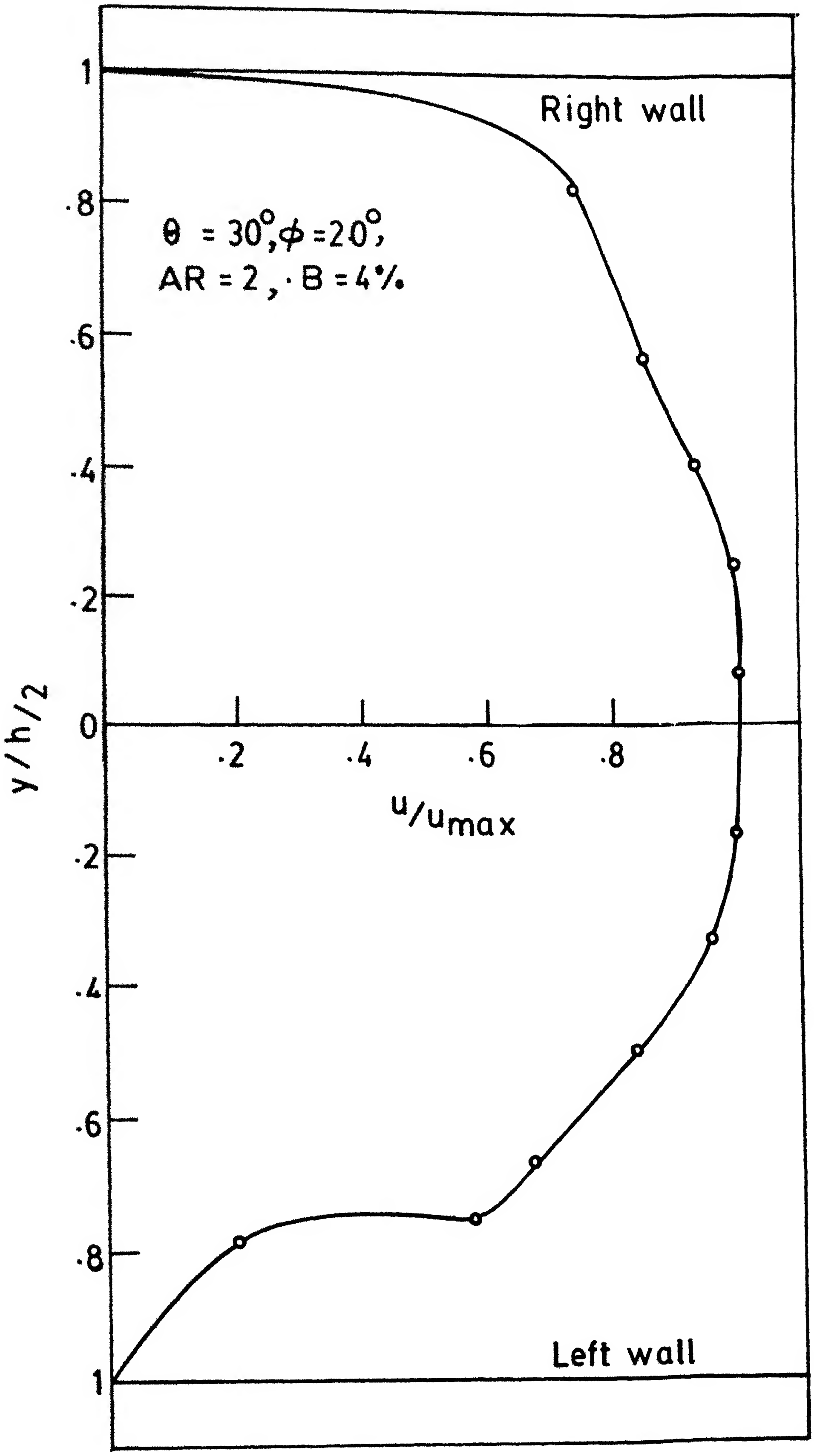


FIG. 19 f EXIT VELOCITY PROFILE ALONG THE HORIZONTAL AXIS TAKEN AT A DISTANCE h/4 FROM THE BOTTOM WALL

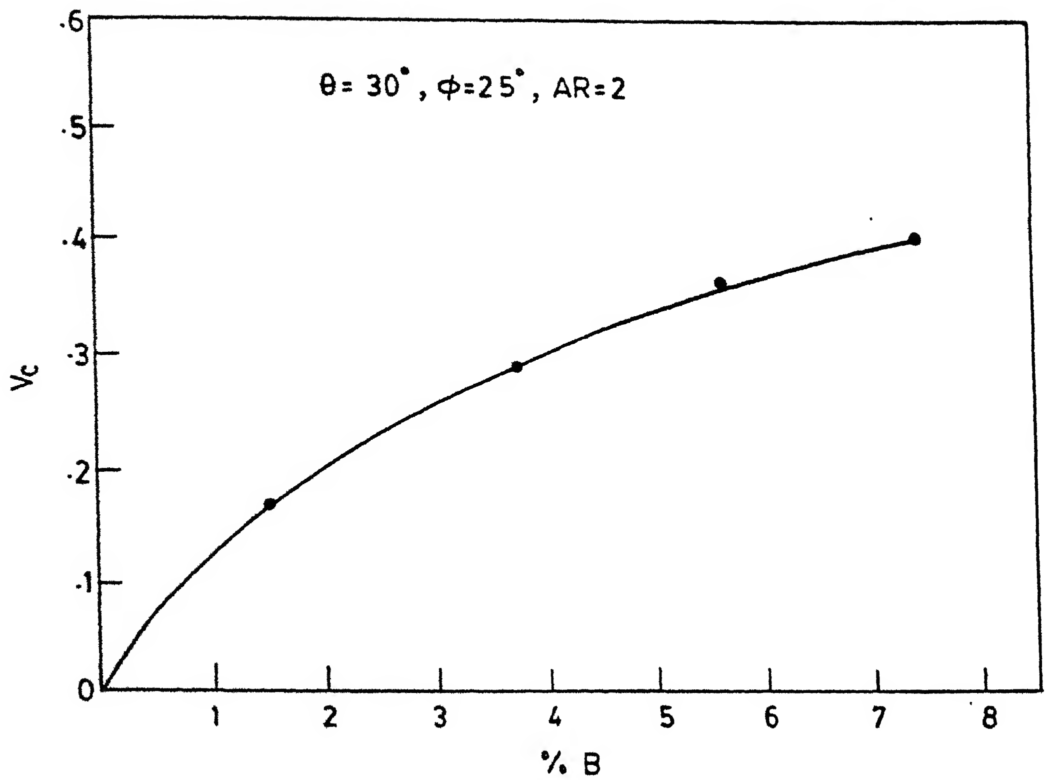


FIG. 20 EFFECT OF SUCTION ON VORTEX CHAMBER DEPRESSION

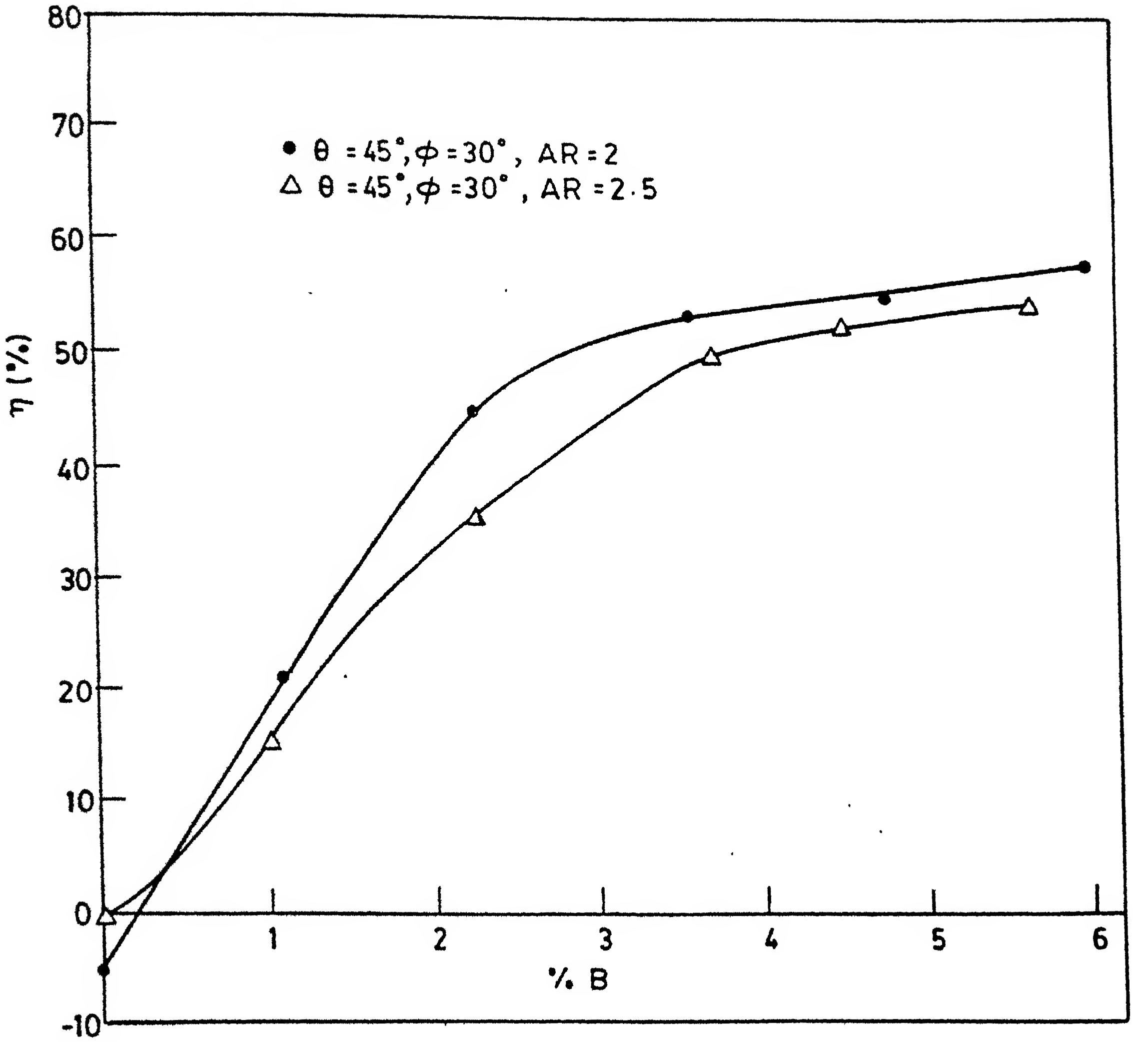


FIG. 21 COMPARISON OF DIFFUSER EFFECTIVENESS FOR TWO AREA RATIOS

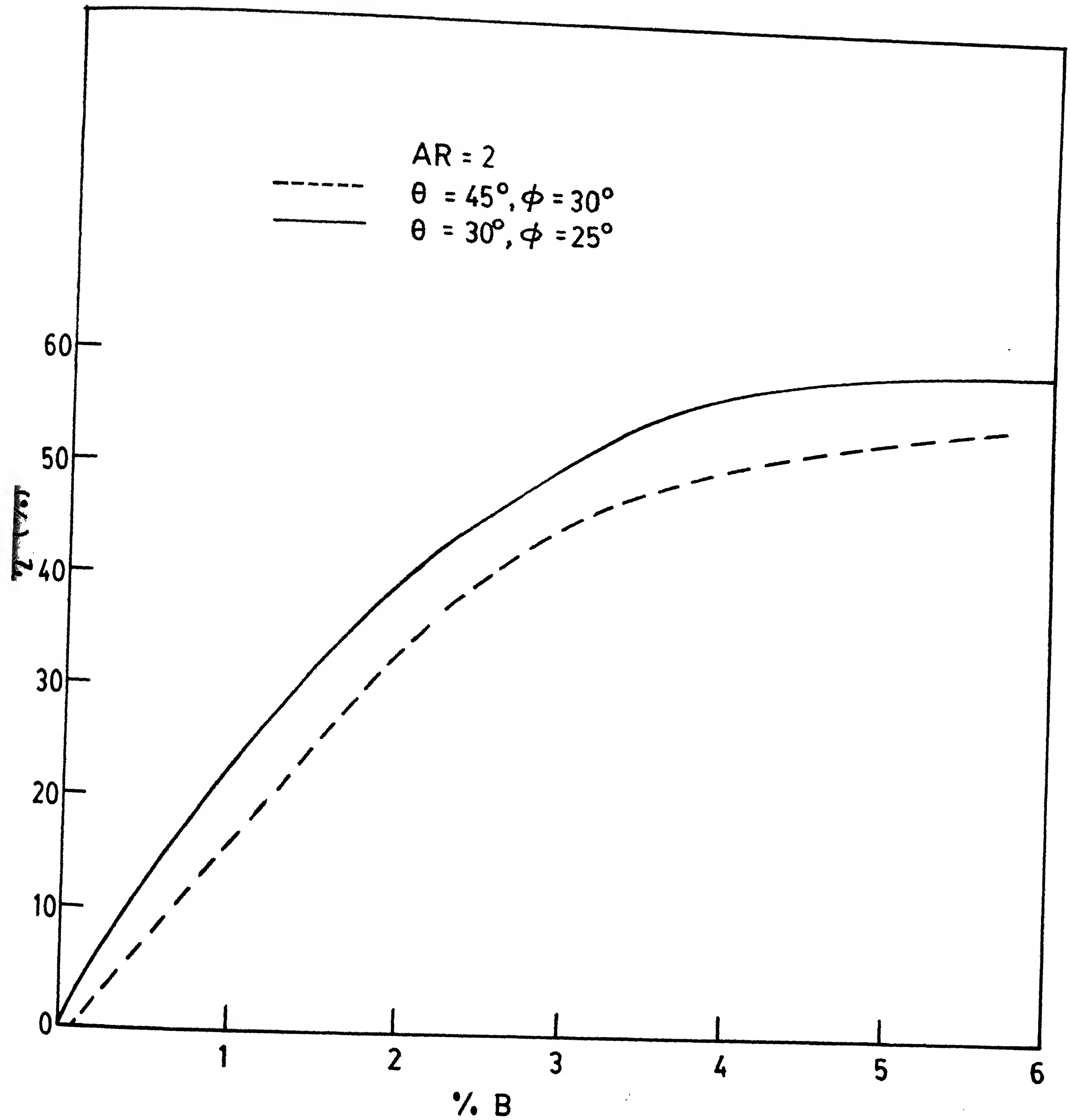


FIG. 22 COMPARISON OF DIFFUSER EFFECTIVENESS FOR 2 DIVERG ENCE ANGLES

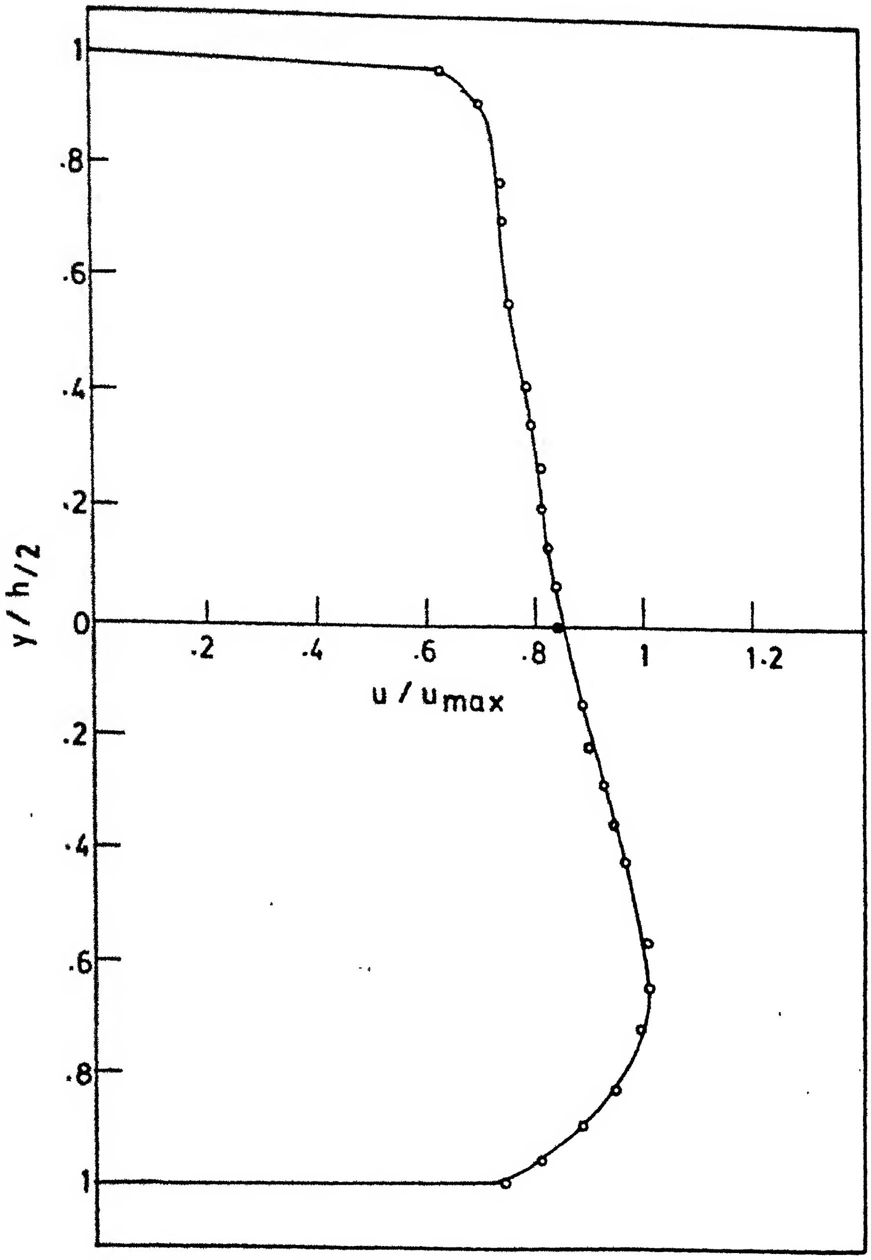


FIG. 23b VELOCITY PROFILE ALONG THE VERTICAL AXIS AT THE EXIT OF PRIMARY DUCT

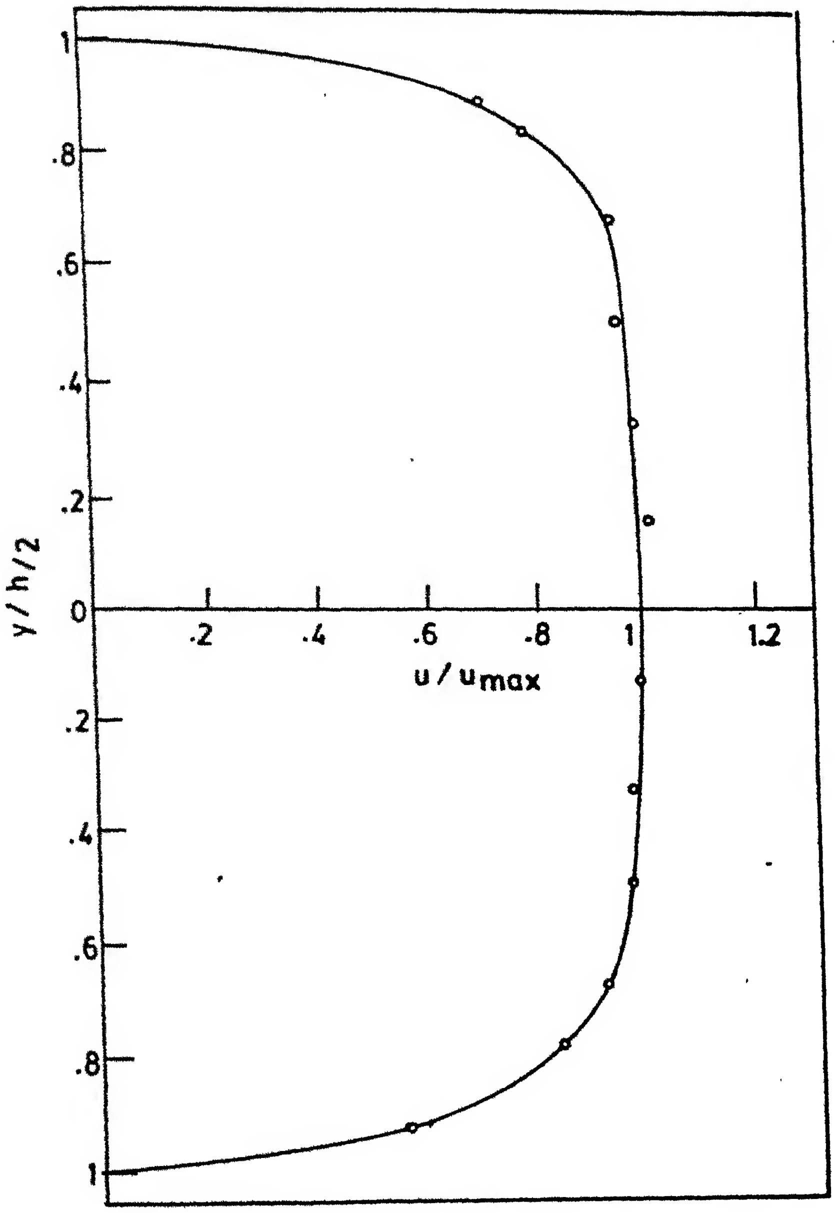


FIG. 23b VELOCITY PROFILE ALONG THE HORIZONTAL AXIS AT THE EXIT OF PRIMARY DUCT

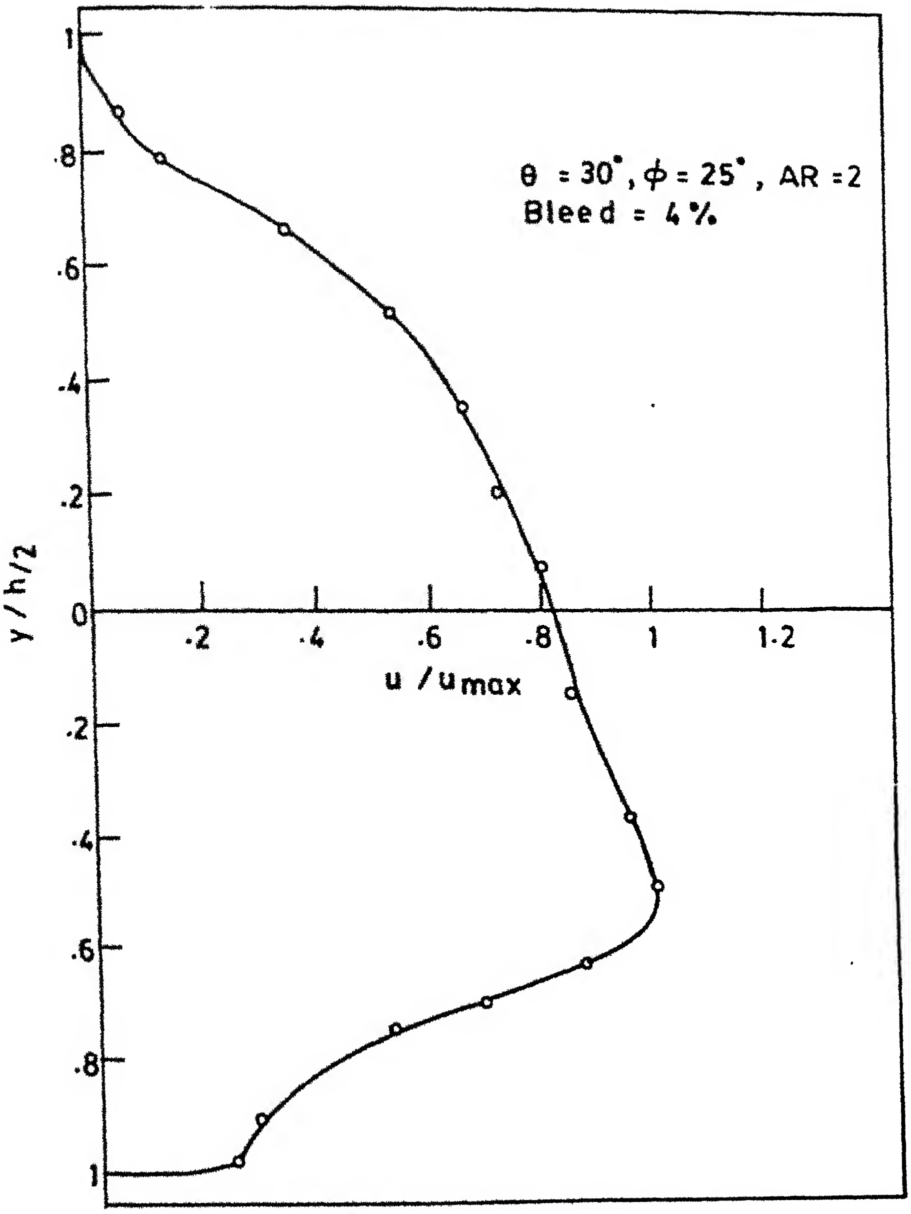


FIG. 24 DIFFUSER EXIT VELOCITY PROFILE WITH SUCTION FROM BOTH SIDES

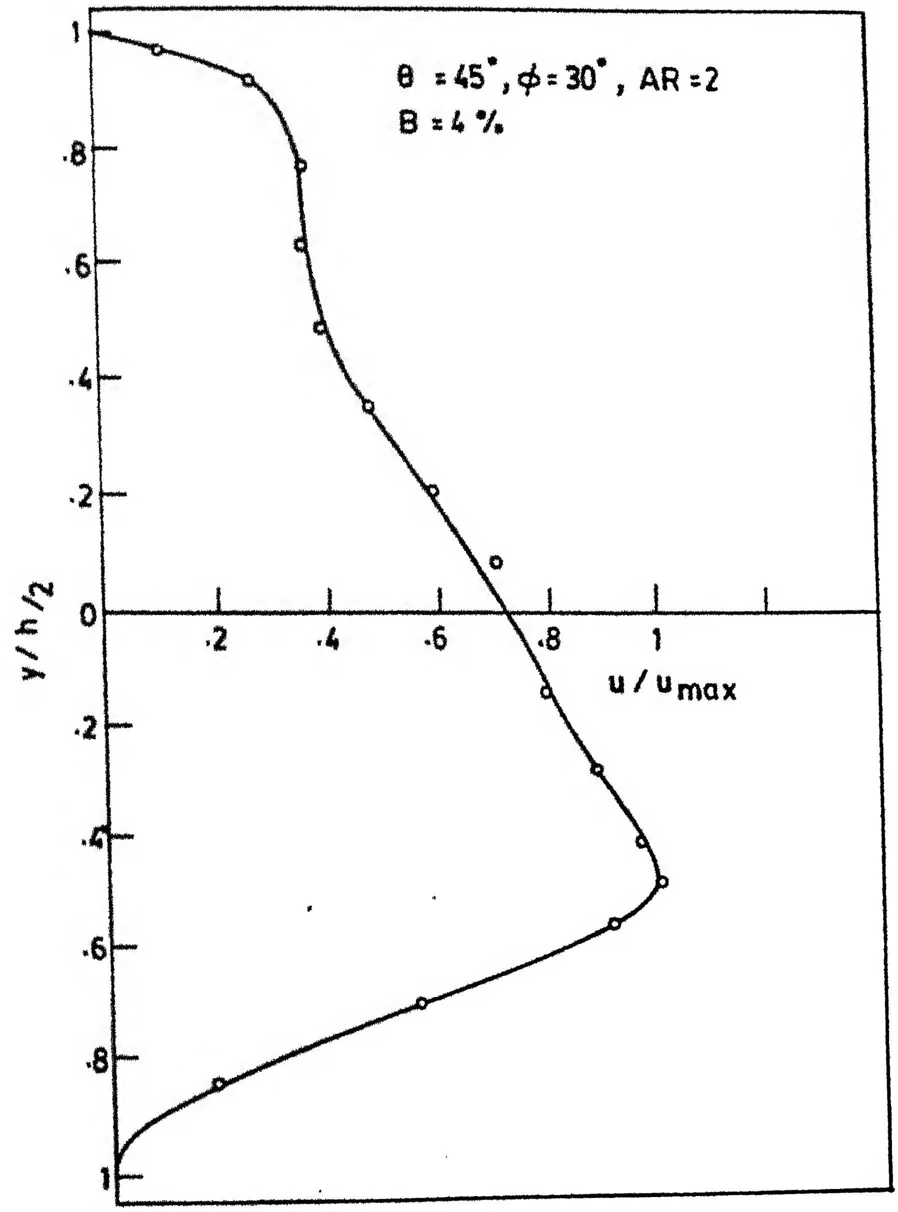


FIG. 25 VELOCITY PROFILE AT THE DIFFUSER EXIT WITH BOTH SIDE SUCTION

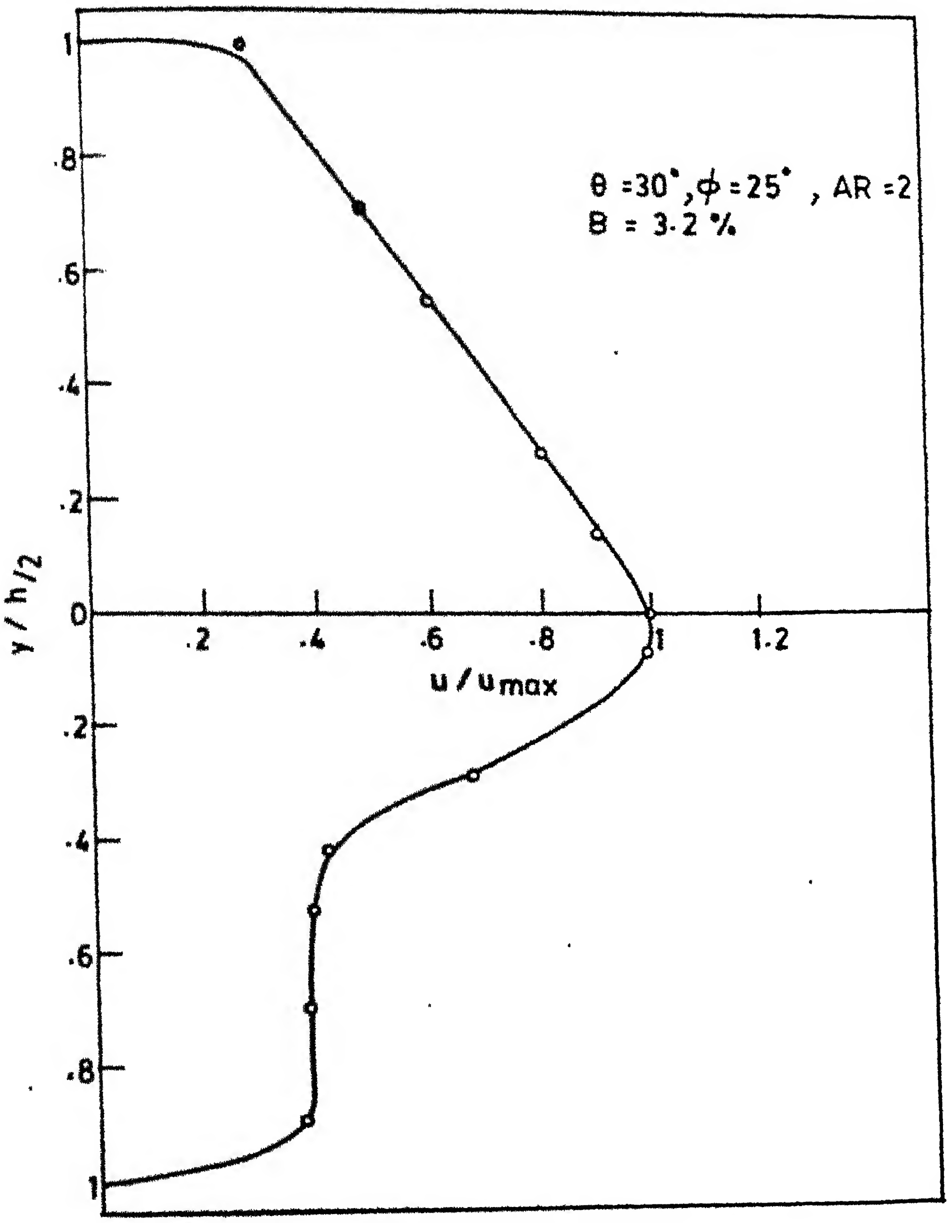


FIG. 26 VELOCITY PROFILE WITH SUCTION ONLY ON TOP WALL

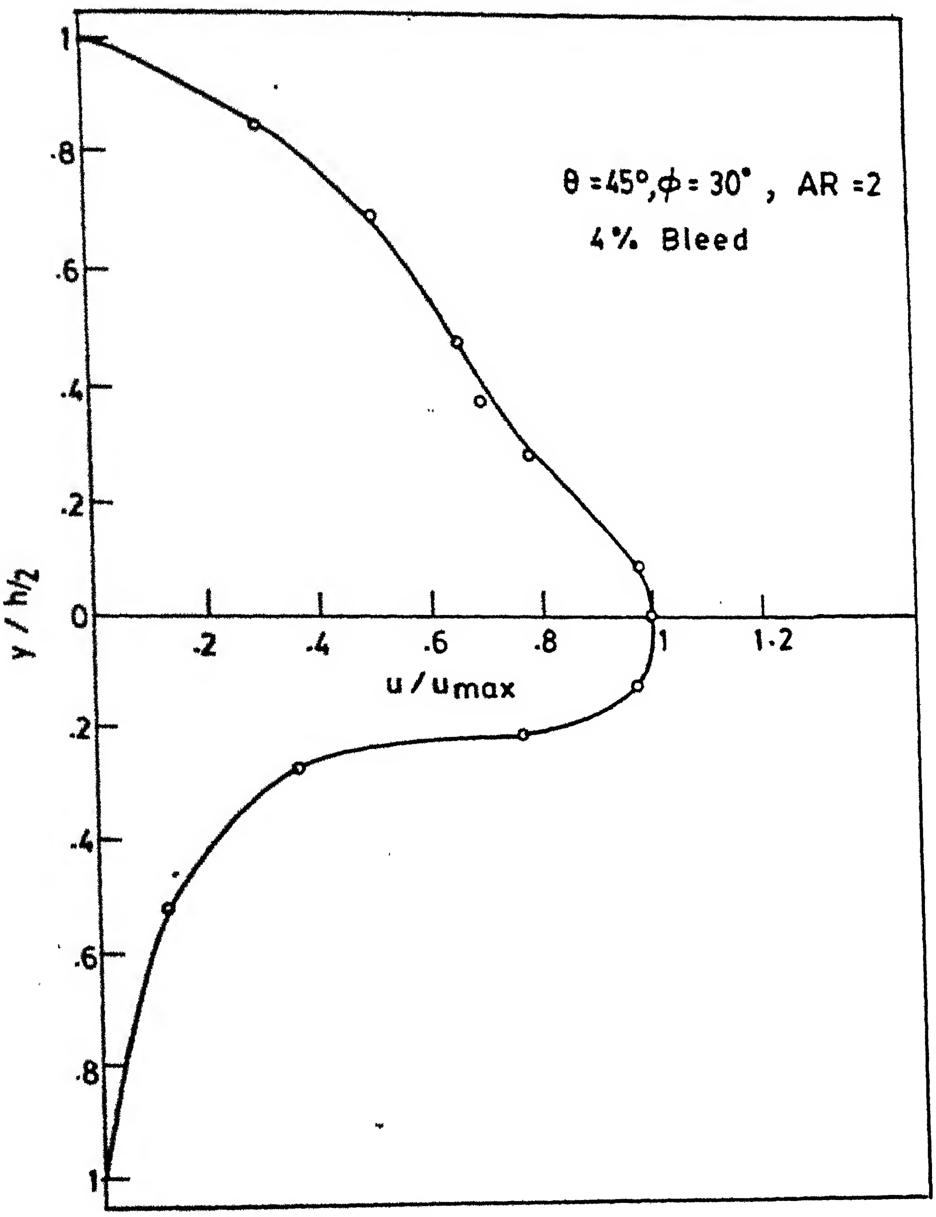


FIG. 27 DIFFUSER EXIT VELOCITY PROFILE WITH SUCTION ONLY FROM TOP WALL

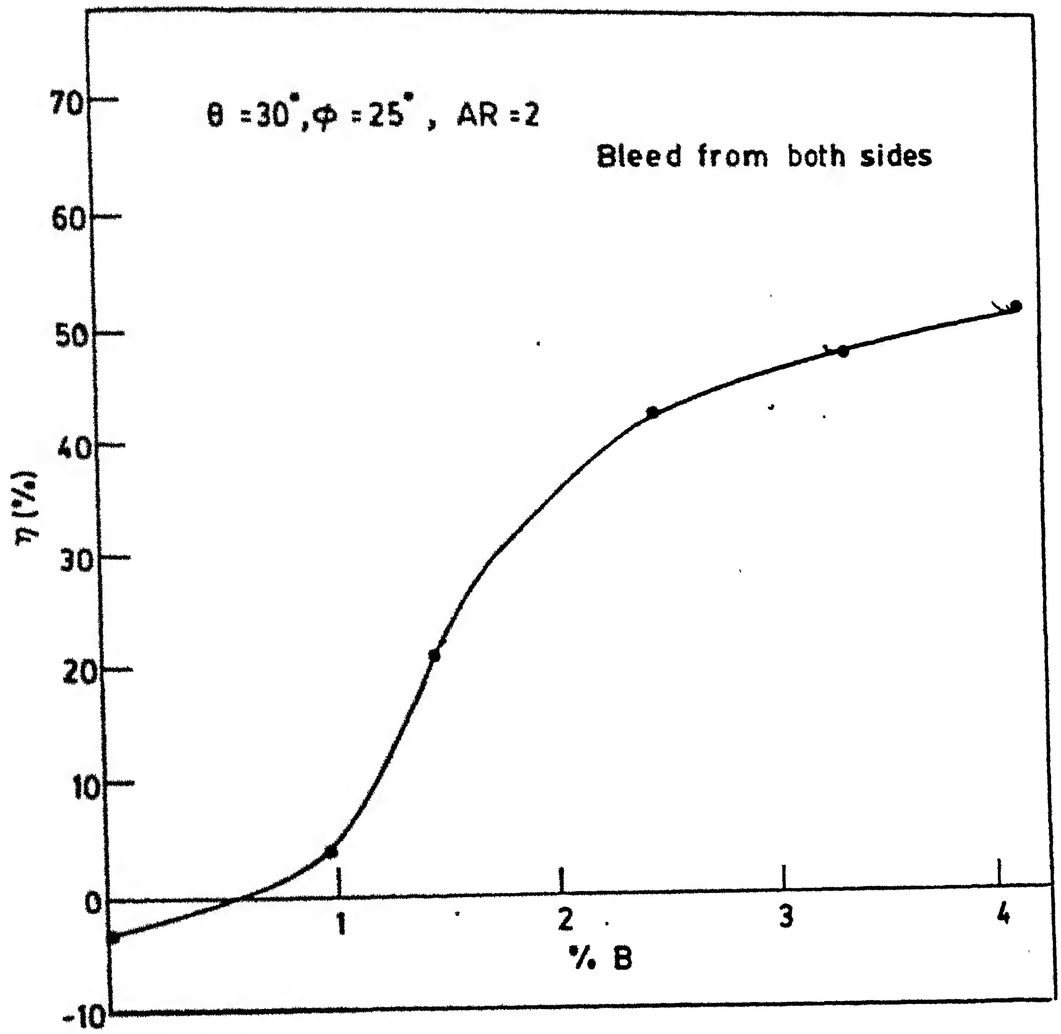


FIG. 28 EFFECT OF SUCTION ON DIFFUSER EFFECTIVENESS

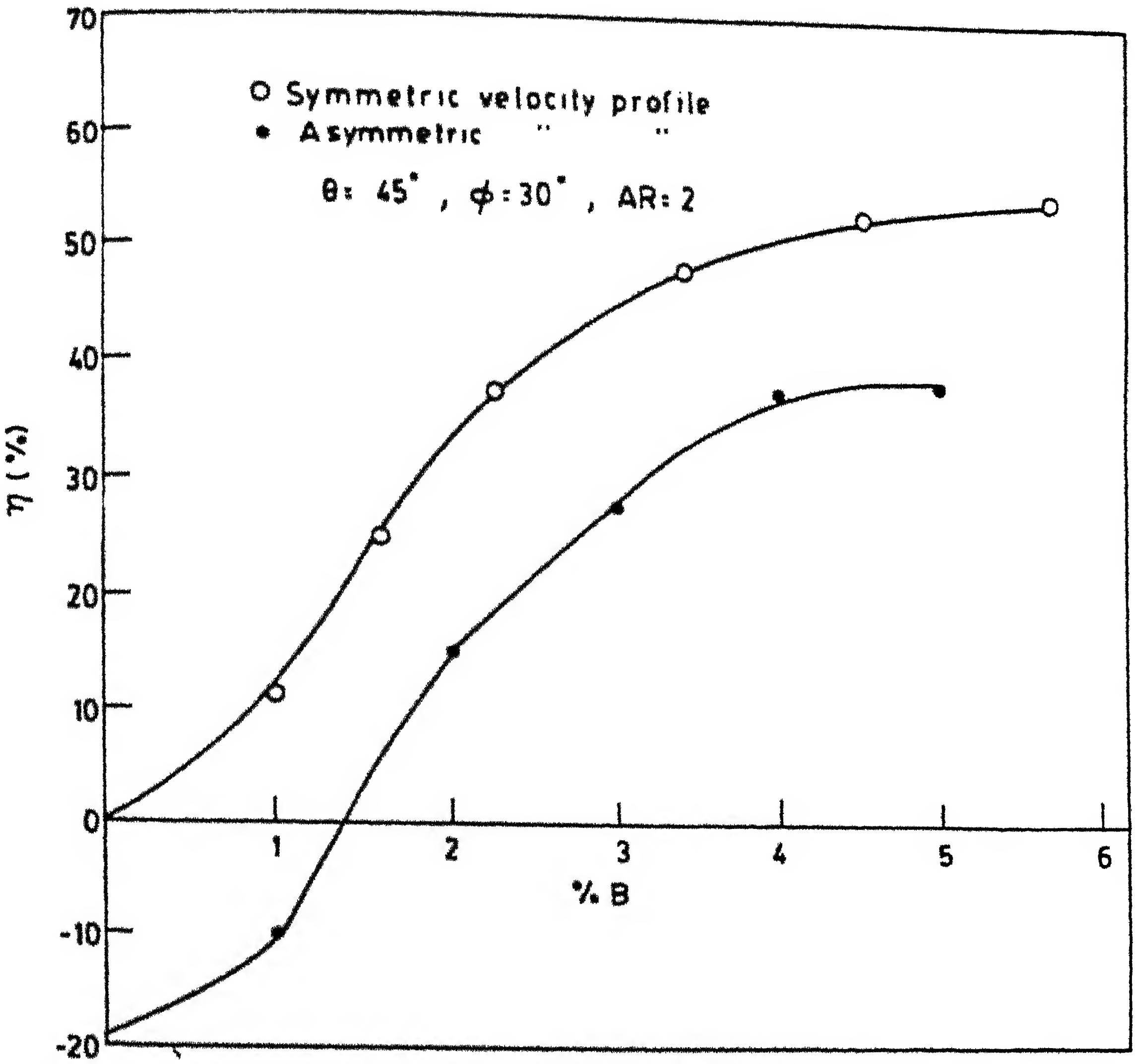


FIG. 29 EFFECT OF ASYMMETRY OF VELOCITY PROFILE ON PRESSURE RECOVERY

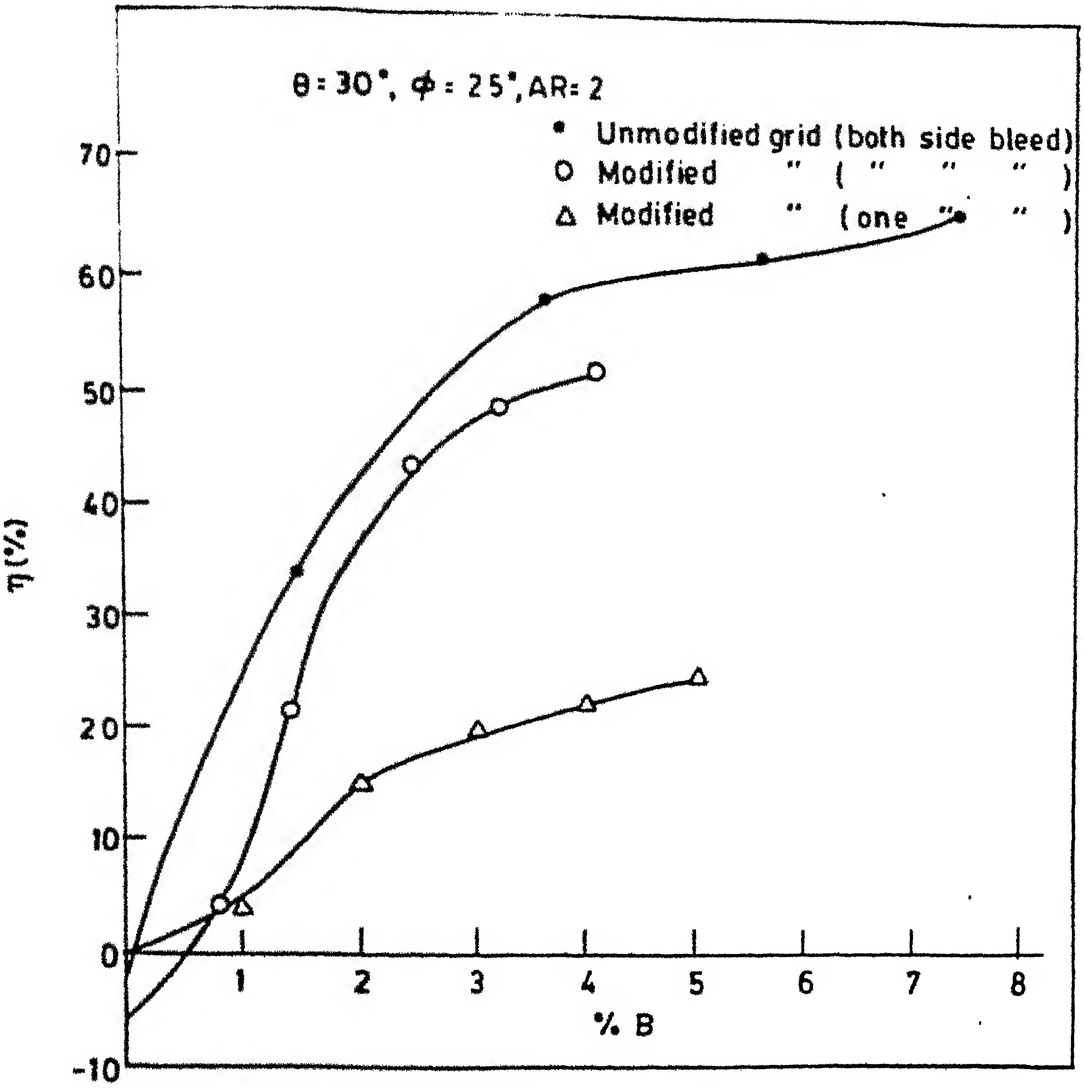


FIG. 30 EFFECT OF INLET VELOCITY PROFILE ON DIFFUSER EFFECTIVENESS

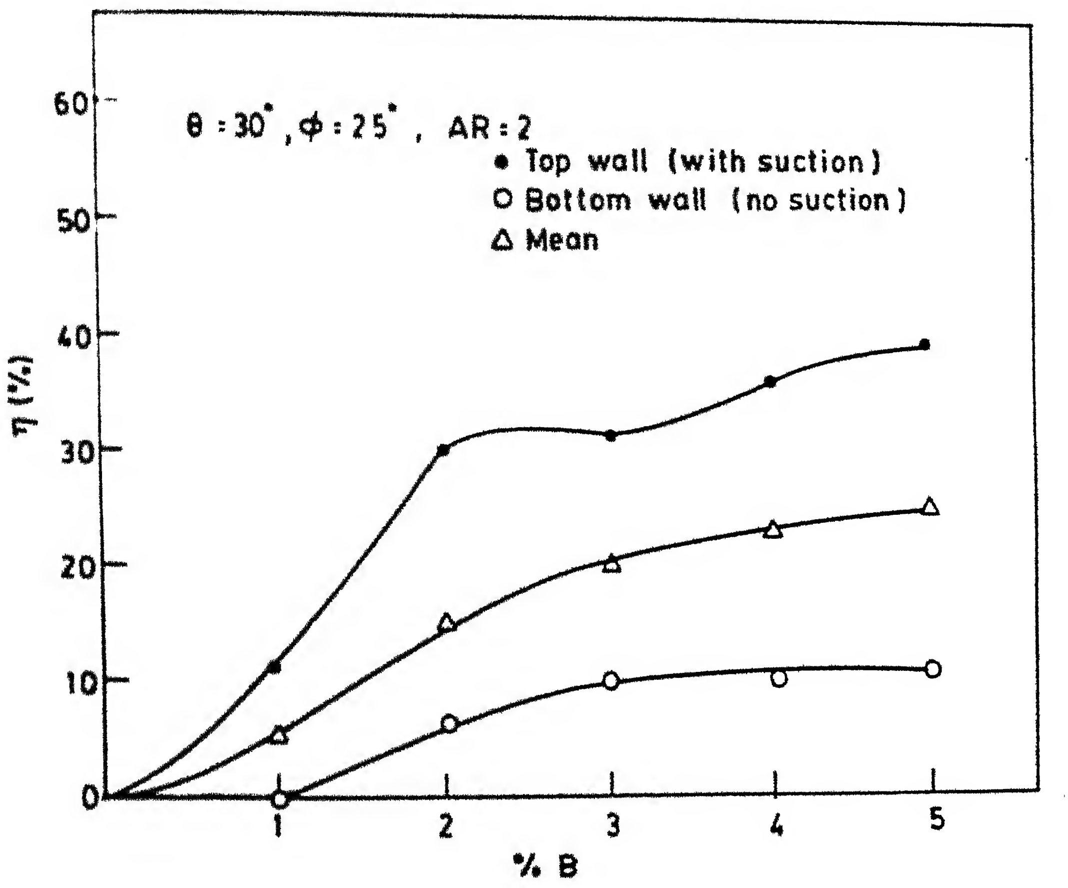
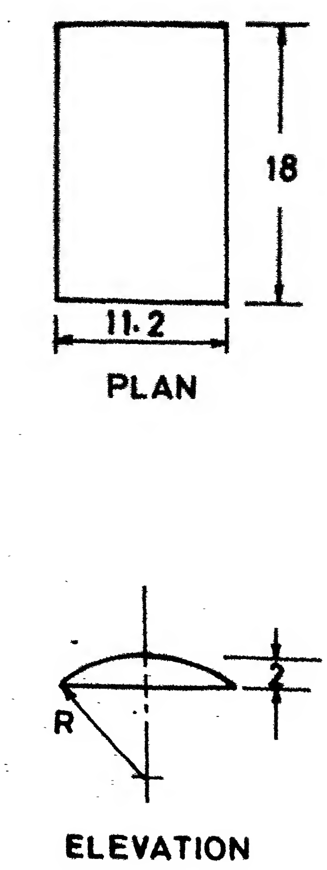


FIG 31 EFFECT OF SUCTION ON DIFFUSER EFFECTIVENESS



VORTEX GENERATOR

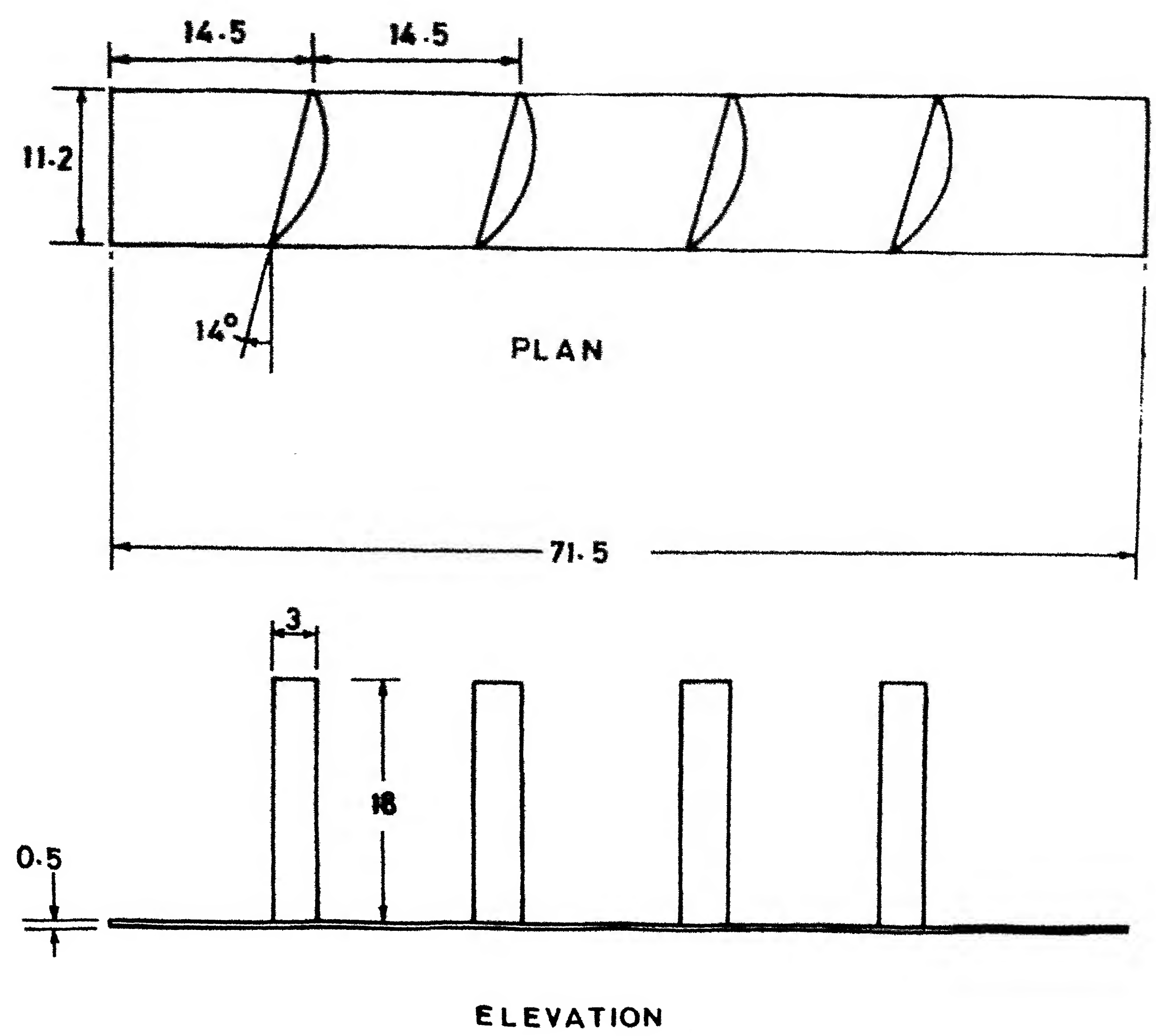


FIG.32a VORTEX GENERATOR ASSEMBLY (2-off)

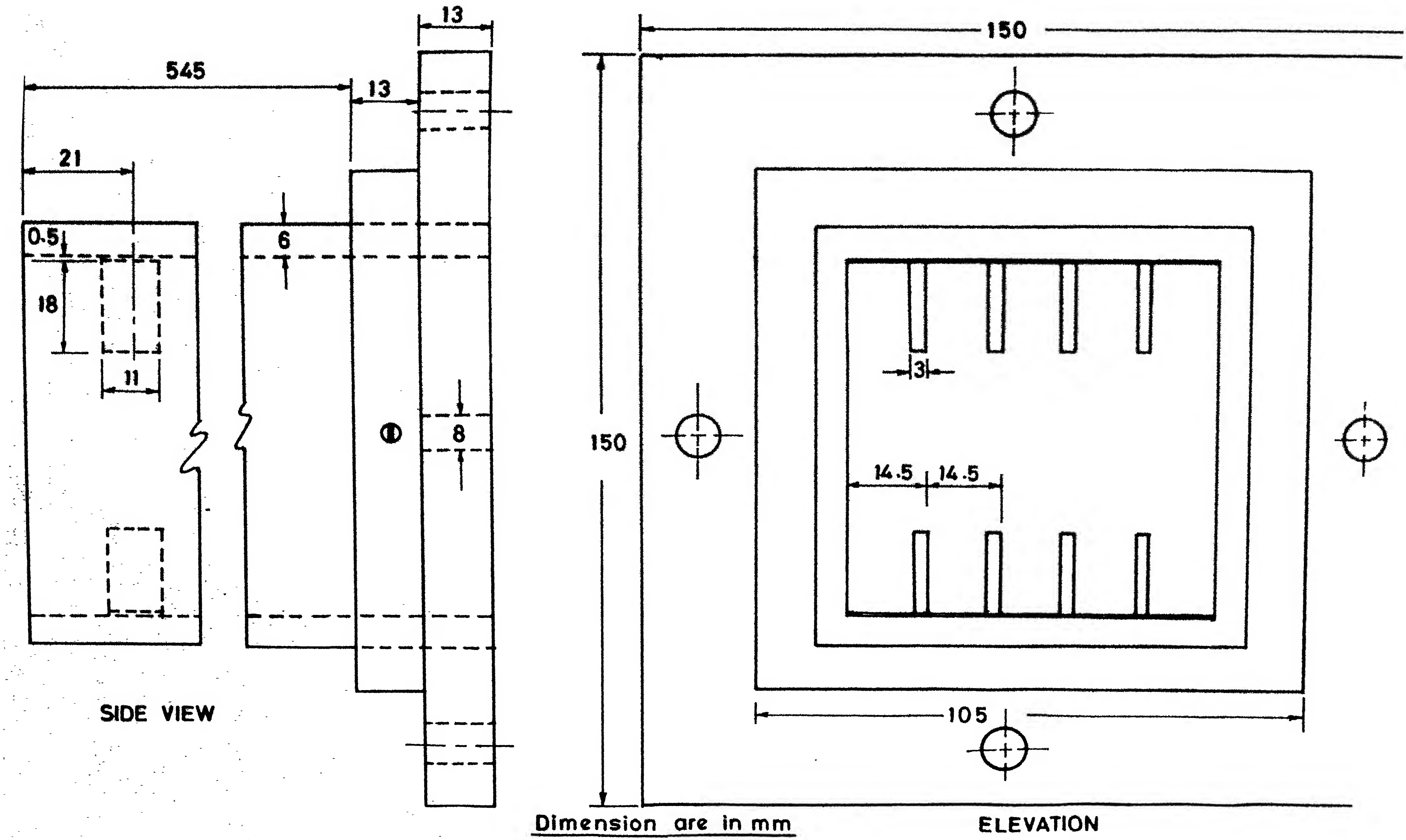


FIG. 32b LOCATION OF VORTEX GENERATES IN THE PRIMARY DUCT

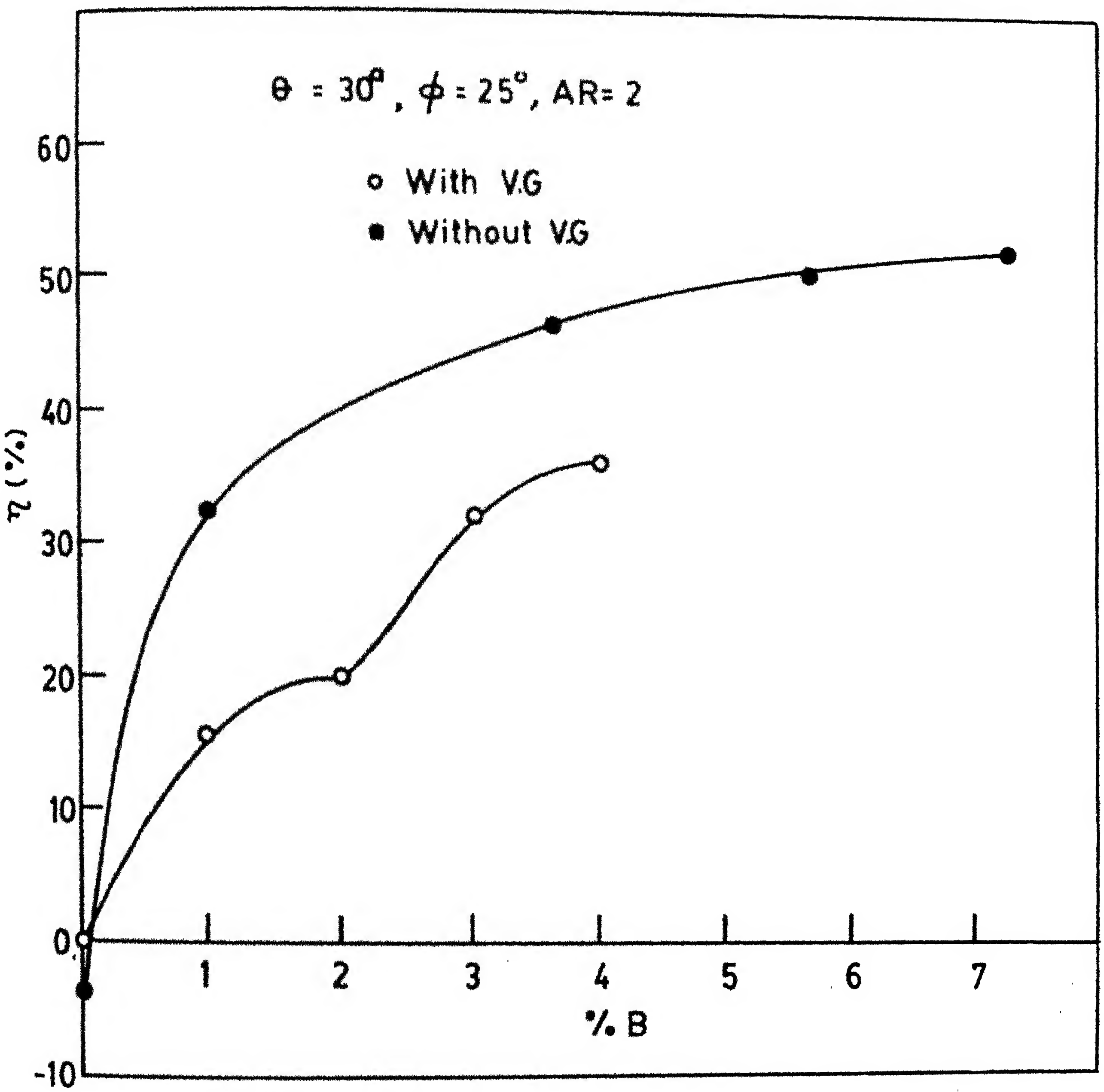


FIG. 33 COMPARISON OF DIFFUSER EFFECTIVENESS WITH AND WITHOUT VORTEX GENERATORS FOR VARIOUS BLEED RATES.

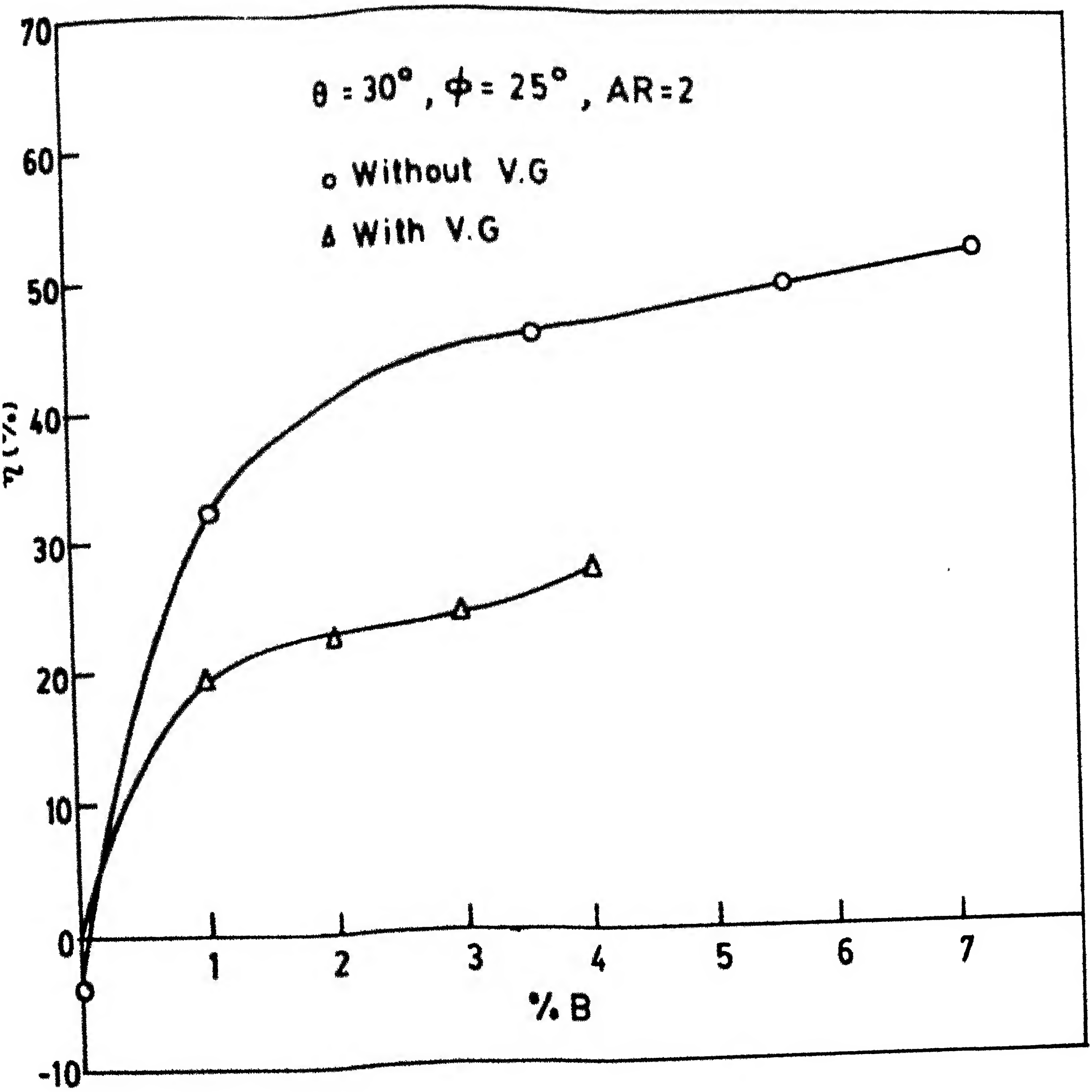


FIG. 34 COMPARISON OF DIFFUSER EFFECTIVENESS WITH AND WITHOUT VORTEX GENERATES

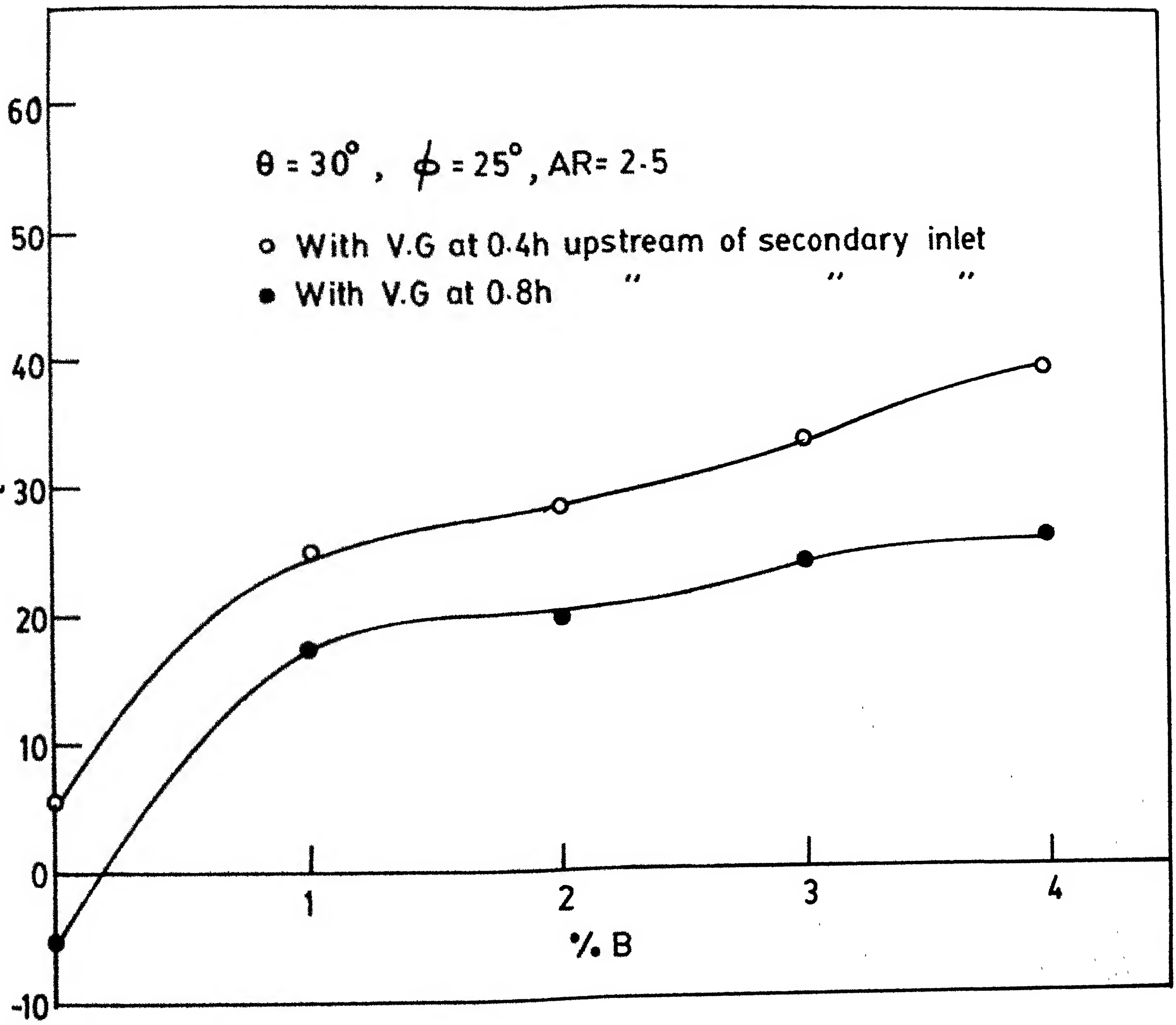


FIG. 35 EFFECT OF VORTEX GENERATOR LOCATION ON DIFFUSER EFFECTIVENESS